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LIQUID ROCKET ENGINE TURBOPUMP BEARINGS

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FOREWORD

NASA experience has indicated a need for uniform criteria for the design of space vehicles. Accordingly, criteria are being developed in the following areas of technology:

Environment
Structures
Guidance and Control
Chemical Propulsion

Individual components of this work will be issued as separate monographs as soon as they are completed. This document, part of the series on Chemical Propulsion, is one such monograph. A list of all monographs issued prior to this one can be found on the last page of this document.

These monographs are to be regarded as guides to design and not as NASA requirements, except as may be specified in formal project specifications. It is expected, however, that these documents, revised as experience may indicate to be desirable, eventually will provide uniform design practices for NASA space vehicles.

This monograph, "Liquid Rocket Engine Turbopump Bearings," was prepared under the direction of Howard W. Douglass, Chief, Design Criteria Office, Lewis Research Center; project management was by Harold W. Schmidt and Lionel Levinson. The monograph was written by Myles F. Butner of Rocketdyne Division, North American Rockwell Corporation, and was edited by Russell B. Keller, Jr. of Lewis. To assure technical accuracy of this document, scientists and engineers throughout the technical community participated in interviews, consultations, and critical review of the text. In particular, Fred R. Mallaire of Aerojet-General Corporation, Robert L. Thomas of Pratt and Whitney Aircraft Division, United Aircraft Corporation, and William J. Anderson and Herbert W. Scibbe of the Lewis Research Center individually and collectively reviewed the text in detail.

Comments concerning the technical content of this monograph will be welcomed by the National Aeronautics and Space Administration, Lewis Research Center (Design Criteria Office), Cleveland, Ohio 44135.

March 1971

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GUIDE TO THE USE OF THIS MONOGRAPH

The purpose of this monograph is to organize and present, for effective use in design, the significant experience and knowledge accumulated in development and operational programs to date. It reviews and assesses current design practices, and from them establishes firm guidance for achieving greater consistency in design, increased reliability in the end product, and greater efficiency in the design effort. The monograph is organized into two major sections that are preceded by a brief introduction and complemented by a set of references.

The State of the Art, section 2, reviews and discusses the total design problem, and identifies which design elements are involved in successful design. It describes succinctly the current technology pertaining to these elements. When detailed information is required, the best available references are cited. This section serves as a survey of the subject that provides background material and prepares a proper technological base for the *Design Criteria* and Recommended Practices.

The *Design Criteria*, shown in italic in section 3, state clearly and briefly what rule, guide, limitation, or standard must be imposed on each essential design element to assure successful design. The *Design Criteria* can serve effectively as a checklist of rules for the project manager to use in guiding a design or in assessing its adequacy.

The Recommended Practices, also in section 3, state how to satisfy each of the criteria. Whenever possible, the best procedure is described; when this cannot be done concisely, appropriate references are provided. The Recommended Practices, in conjunction with the *Design Criteria*, provide positive guidance to the practicing designer on how to achieve successful design.

Both sections have been organized into decimally numbered subsections so that the subjects within similarly numbered subsections correspond from section to section. The format for the Contents displays this continuity of subject in such a way that a particular aspect of design can be followed through both sections as a discrete subject.

The design criteria monograph is not intended to be a design handbook, a set of specifications, or a design manual. It is a summary and a systematic ordering of the large and loosely organized body of existing successful design techniques and practices. Its value and its merit should be judged on how effectively it makes that material available to and useful to the designer.

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LIQUID ROCKET ENGINE TURBOPUMP BEARINGS

1. INTRODUCTION

Bearing requirements for liquid propellant rocket engine turbopumps have forced advances in a specialized branch of bearing technology to fulfill the special and unusual operating conditions of speed, load, life, and environment. Advances in technology presently are being pursued to develop longer life bearings, roller bearings cooled by propellants, higher speed bearings, bearings for service with liquid fluorine, and solution of ball speed variation problems. Also in progress are bearing-material development programs for space environments, applications with heat soakback, hot restarts, and extended exposures to propellants at elevated temperatures.

These advances in bearing technology rely heavily on empirical results of testing under actual or simulated operating environments. In addition, computer calculations are utilized extensively to evaluate the effects of speed, geometry, and loading on bearing life, stresses, and deflections, and to determine bearing spring rates used for shaft dynamics calculations.

Presently, rolling contact bearings are utilized almost exclusively in preference to fluid film bearings because of the following characteristics:

- Large capacity-to-volume ratio
- Ability to operate independently of external pressurizing systems
- Ability to operate satisfactorily after ingesting foreign material
- Tolerance for short periods of coolant/lubrication starvation
- High radial spring rate
- Low heat generation and coolant/lubrication consumption

The development of bearings satisfactory for use with rocket engine turbopumps has depended on the solution of a number of basic problems in bearing design:

(1) Use of nonlubricants for cooling high-speed, high-load bearings has required specialized bearing geometry, careful material selection, adequate coolant quantity, and control of manufacturing processes.

(2) Maintaining quality control of bearings has required adequate specifications for physical properties of what may often be new materials, for material identification, for confirmation of material integrity, for cleaning and packaging, and for inspection of new bearing physical dimensions.

(3) Obtaining shaft dynamic response through design control of the bearing stiffness has required that bearing detail design be performed as part of the basic turbo-

pump layout selection input. Computerized analyses of bearings and shaft systems make possible the rapid evaluation of trial designs.

(4) Permanent race and rolling element dimensional changes that occur when bearings are chilled to cryogenic temperatures have been eliminated by the use of stabilizing chill cycles prior to final machining.
(5) Rolling element size changes due to thermal effects or wear remain a problem. There is some experimental evidence that race relief (unloading chute) may be a solution for ball bearings.
(6) Cage wear problems largely have been overcome by the use of low-friction materials to provide the lubricating function, use of adequate clearances, use of sufficient web thickness to allow for wear, material properties control, and manufacturing quality control.
(7) Damage and wear caused by skidding of rolling elements operating at high speed and low load have been controlled by the use of suitable axial or radial preloading.

Prevention of bearing problems is more effective and less costly than their cure. It is essential that the bearing designer participate in turbopump preliminary design by aiding the mechanical designer in basic decisions that influence the bearing application and overall reliability. In this capacity, the bearing designer should

- Reduce the DN value of the bearings to a minimum
- Minimize the number of bearings used
- Guide the mechanical designer in selecting the proper bearing type
- Secure hydraulic balancing of impeller forces
- Select bearing materials with appropriate chemical and mechanical properties
- Provide for mechanical balancing
- Provide adequate clean coolant supply
- Assure practical means for bearing assembly
- Avoid design practices that lead to severe misalignment and the detrimental combination of radial and thrust load

These provisions are more easily incorporated as integral design features before the design is hardened than as adjustments after detailing is underway.

After a firm turbopump layout has been adopted, the bearing designer should review design details of pump parts affecting bearings, e.g., fits, finishes, and race clamping; perform detailed examination of specimens from bearing tests to detect any need for bearing design improvements; and examine bearings from component (turbopump hot fire) and engine tests to verify that the bearing design is adequate for the actual operating conditions of load, alignment, and cooling. This step is mandatory because accurate prediction of pump loads and temperatures is extremely difficult.

The material that follows has been divided into two categories: (1) the fundamental problems involved in bearing assembly design; and (2) the problems involved in the design of the three basic bearing components—the rolling element, the race, and the cage.

These problems are considered in the order and manner in which the designer must handle them. The designer must first choose a bearing assembly design that will meet all the needs of the proposed application and then design the components to meet the needs of the assembly. In all cases, there is cross-feed between these two design efforts, and accordingly this monograph frequently cross-references and dovetails related material.

2. STATE OF THE ART

2.1 Bearing Assembly Design

Turbopump shafts are generally supported at two bearing positions. These are placed as closely as possible to major rotating components to reduce radial movement and to control critical speeds. Ball bearings (Conrad-type, angular-contact, and split-inner-ring) and cylindrical roller bearings are used because of their capabilities in load, speed, stiffness, and misalignment tolerances (see table I). Film bearings have found limited use in specialized applications. Tapered roller bearings, needle bearings, and pure-

TABLE I.—Comparison of Turbopump Bearings

Bearing type	Advantages	Disadvantages	Primary condition for use
Conrad-type ball	Any combination of radial and thrust direction; large misalignment capability; moment load capacity	Limited number of balls; two-piece cage necessary	Combined load; two-direction thrust loads
Angular-contact ball	Thirty percent more capacity than similar size Conrad; one-piece cage	Predominant thrust required; one-direction capacity; lower misalignment tolerances than Conrad	High speed, high load; single-direction thrust
Split-ring ball	Thirty percent more thrust capacity than similar size Conrad; one-piece cage; two-direction thrust capability; lower axial clearance through use of gothic arch	Predominant thrust required; lower misalignment tolerance than Conrad	Two-direction thrust
Cylindrical roller	Much higher radial capacity than ball bearing; provides axial freedom of shaft; higher radial stiffness than ball bearings; one-piece cage	No axial load capacity; roller ends wear in nonlubricating coolants; lower misalignment tolerance than ball bearings	High radial capacity without axial restraint; high radial stiffness
Balance piston	High thrust capacity	Must be protected from thrust loads during start; has no capacity without pressure; must be carefully designed for stable operation	Pump shaft thrust exceeds rolling bearing capacity

thrust ball bearings have not been used in rocket engine turbopumps because of their limited speed capabilities.

Table II presents a summary of current turbopump bearing practice and state of the art regarding bearing type, size, speed, load, coolant use, and other factors relevant to displaying bearing capabilities.

The first consideration in the design or selection of a bearing is its directional loading capability for the intended application. Then, for bearings that are to be used at high speeds, relationships of speed-load-life that account for centrifugal force must be used. Further, because bearing radial stiffness is a controlling factor in critical speed control, it is often a primary consideration in bearing type selection and bearing location on the turbopump shaft.

2.1.1 Load Capability

Currently, Conrad-type ball bearings are often used because of their ability to support a combined radial and axial load, a thrust load in both axial directions, and moment loading. Their capacity depends on the number of balls, which is limited by the assembly method. Also, they require a two-piece cage, which limits their speed capability.

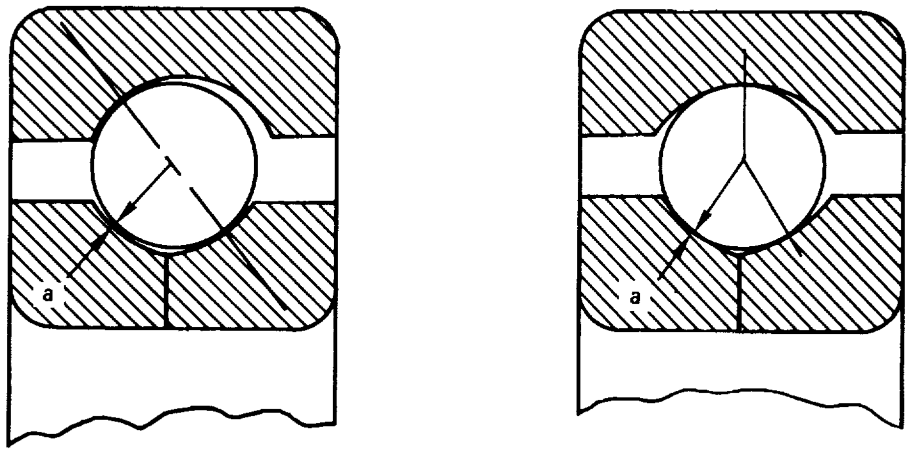
Split-inner-ring ball bearings and angular-contact bearings, which have approximately 30 percent more thrust capacity than Conrad-type bearings, are used singly or in tandem to support heavy axial loads. The advantages offered by split-inner-ring bearings are the use of a one-piece cage, two-way thrust load capability, and the ability to lower the total end play of the bearing through the use of the gothic arch inner race. This latter feature makes a predominant thrust load requirement for satisfactory operation. A predominant radial load results in a change from a normal operation (fig. 1(a)) to a three-point contact (fig. 1(b)), and consequent race-ball distress. Design considerations are given in reference 1, pp. 44 and 45. Angular-contact ball bearings are used only for predominantly axial loads so that ball contact with the low shoulder does not occur (fig. 2(a)). A conservative rule of thumb for split-ring and angular-contact bearings is to ensure that the thrust load is always twice the radial load.

Tandem duplexed ball bearings are used when the thrust load is greater than the capability of a single bearing with ball size maximized to centrifugal limits. Two or three bearings are used to share the loading (ref. 2). To ensure load sharing, face flushness, bore and outside diameters, and contact angles are specified within close limits in each set of bearings; selective assembly may be used rather than extremely tight tolerances. Duplexed ball bearings with opposing contact angles are used to limit axial deflection and to obtain increases in bearing radial spring rate (figs. 2(b) and 2(c)). Preloaded springs are often used to avoid thermal loading of the bearings.

TABLE II.–Summary of Current Turbopump Bearing Capabilities

Bearing type	Bore, mm	Materials			Load, lb_f		Speed, 10^3 rpm	Coolant		Experience		Maximum test hours per bearing (no failure)	Maximum temperature, °F
		Ball or roller	Races	Cage	Axial	Radial		Fluid	Flowrate, gpm	Turbopump	Test rig		
Ball	35	Halmo	Halmo	S-Monel	1,400	1350	40	RP-1	0.2	X		5	800
Ball	70	52100	52100	AMS 4616	5,900	4700	6.8	RP-1	0.1	X		1.75	250
Tandem Ball	130	52100	52100	AMS 4616	55,000	0	5.5	RP-1	4.0	X		20	350
Roller	35	4620	4620	AMS 4616	0	3700	40.0	RP-1	0.2	X		5	350
Roller	70	52100	52100	AMS 4616	0	4000	6.8	RP-1	0.1	X		1.75	250
Roller	170	[11]	[11]	AMS 4616	0	500	5.5	RP-1	1	X		100	800
Ball	40	K5H[1]	440-C	25% GFT[2]	2,700	NA	40	50-50[3]	14		X	0.2	NA
Ball	45	K-96[4]	K-96	Graphite[12]	300	0	25	50-50	1.1		X	0.6	NA
Roller	40	440-C	440-C	25% GFT	0	500	40	50-50	8 to 10		X	0.2	NA
Ball	45	440-C	440-C	Armalon	330	0	25	EDA[5]	1.5		X	1	NA
Ball	45	440-C	440-C	Armalon	315	0	25	B_5H_9[6]	1		X	0.4	NA
Ball	45	440-C	440-C	Armalon	800	100	28.3	Liquid hydrogen	100	X		6	-350
Ball	45	440-C	440-C	Armalon	300	0	34.5	Liquid hydrogen	10		X	9	NA
Ball	50	440-C	440-C	Armalon	3,000	NA	24	Liquid hydrogen	20	X		3	NA
Ball	60	440-C	440-C	Armalon	800	100	28.3	Liquid hydrogen	20	X		6	-350
Ball	65	440-C	440-C	Armalon	2,200	100	31.5	Liquid hydrogen	40	X		3	-350
3 Tandem Ball	110	440-C	440-C	Armalon	36,000	NA	13.3	Liquid hydrogen	150		X	0.75	NA
Ball	150	440-C	440-C	Armalon	1,000	0	20	Liquid hydrogen	150		X	0.5	NA
Ball	200	440-C	440-C	Armalon	5,000	0	15	Liquid hydrogen	150		X	1.6	NA
Roller	50	440-C	440-C	Armalon	0	2000	24	Liquid hydrogen	20	X		3	NA
Roller	120	440-C	440-C	Armalon	0	5000	13.3	Liquid hydrogen	26		X	1.63	NA
Ball	35	440-C	440-C	Rulon	0 to 400	150	30	Gaseous hydrogen	0.02 to 0.2 lb/sec	X		5.5	-400 to +60
Ball	40	440-C	440-C	Rulon	140	200	30	Gaseous hydrogen	0.03 lb/sec	X		5.5	-400 to +60
Ball	45	440-C	440-C	Armalon	300	0	34.5	Gaseous hydrogen	0.1 lb/sec		X	9	NA
Roller	30	440-C	440-C	[7]	0	300	12	Gaseous hydrogen	NA	X		5.5	-60
Roller	40	440-C	440-C	[7]	0	200	12	Gaseous hydrogen	0.02 lb/sec	X		5.5	-60
Ball	45	440-C	440-C	Armalon	330	0	25	IRFNA[8]	5 gpm		X	1	NA
Ball	40	440-C	440-C	25% GFT	1,500	NA	40	N_2O_4	5 to 15		X	0.25	NA
Ball	45	440-C	440-C	Armalon	330	0	25	N_2O_4	2 to 5		X	1	NA
Ball	50	K5H	440-C	25% GFT	2,500	NA	25	N_2O_4	5 to 15		X	0.25	NA
Roller	40	440-C	440-C	25% GFT	0	1000	40	N_2O_4	5 to 15		X	0.25	NA
Ball	45	440-C	440-C	Armalon	500	0	25	Liquid oxygen	5		X	15	-250
Ball	60	440-C	440-C	Armalon	3,500	0	8.8	Liquid oxygen	10	X		6	-250
Ball	110	440-C	440-C	Armalon	18,000	NA	4	Liquid oxygen	8.5		X	0.9	NA
Roller	105	440-C	440-C	Armalon	0	3750	4	Liquid oxygen	2		X	2	NA
Ball	45	440-C	440-C	K-Monel	400	0	3.6	Liquid fluorine	Submerged		X	1	NA
Ball	45	K162B[9]	440-C	K-Monel	400	0	3.6	Liquid fluorine	Submerged		X	1	NA
Ball	50	440-C	440-C	BN440[10]	400 to 12	0	20	Liquid fluorine	12 to 20		X	0.5	-293

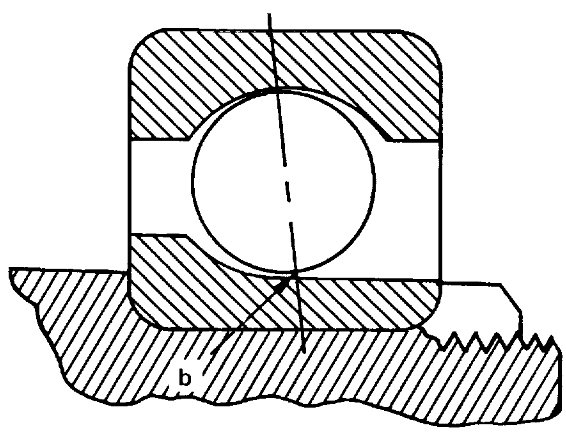
[1]Tungsten-titanium carbide
[2]Glass-filled teflon
[3]UDMH-N_2H_4 (50-50 mixture)
[4]Tungsten carbide
[5]Ethylene diamine
[6]Pentaborane
[7]Plated lead on silver on AMS 4616
[8]Inhibited red fuming nitric acid
[9]Titanium carbide
[10]Berylco Nickel 440
[11]Bower 315
[12]U.S. graphite, grade 2



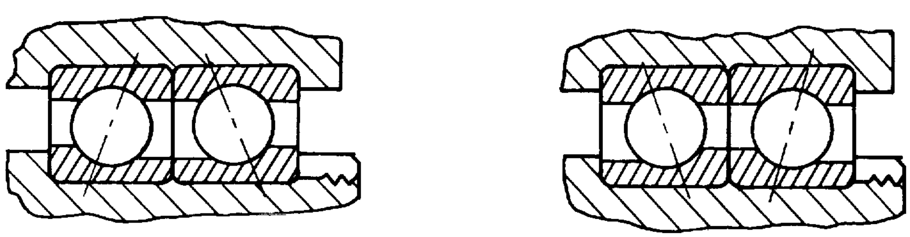
(a) Normal operation
(clearance at a)

(b) Three-point contact
(wiping contact at a)

Figure 1.—Split-inner-ring ball bearing.



(a) Low shoulder contact at b

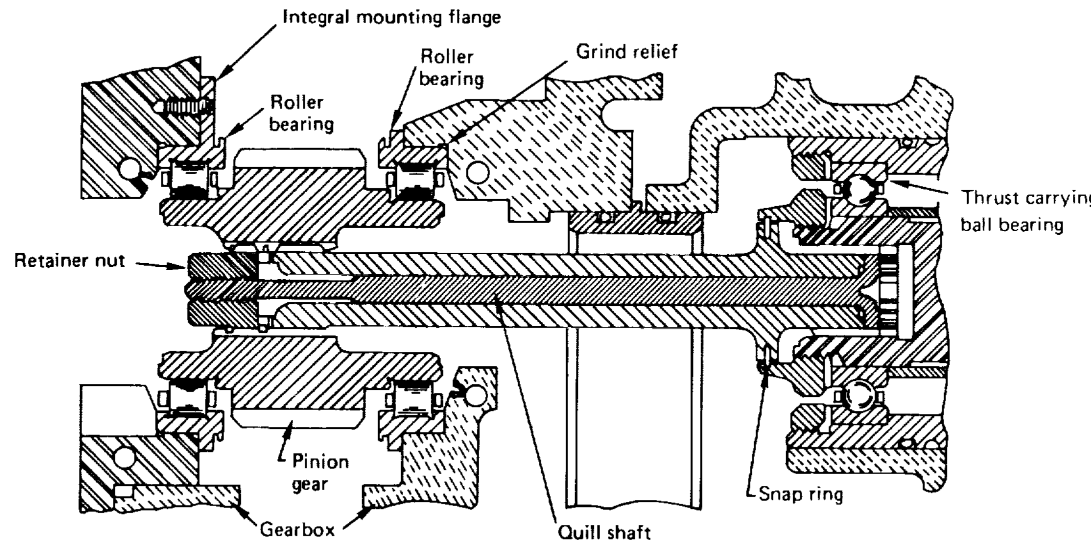


(b) Duplex DB (back-to-back)

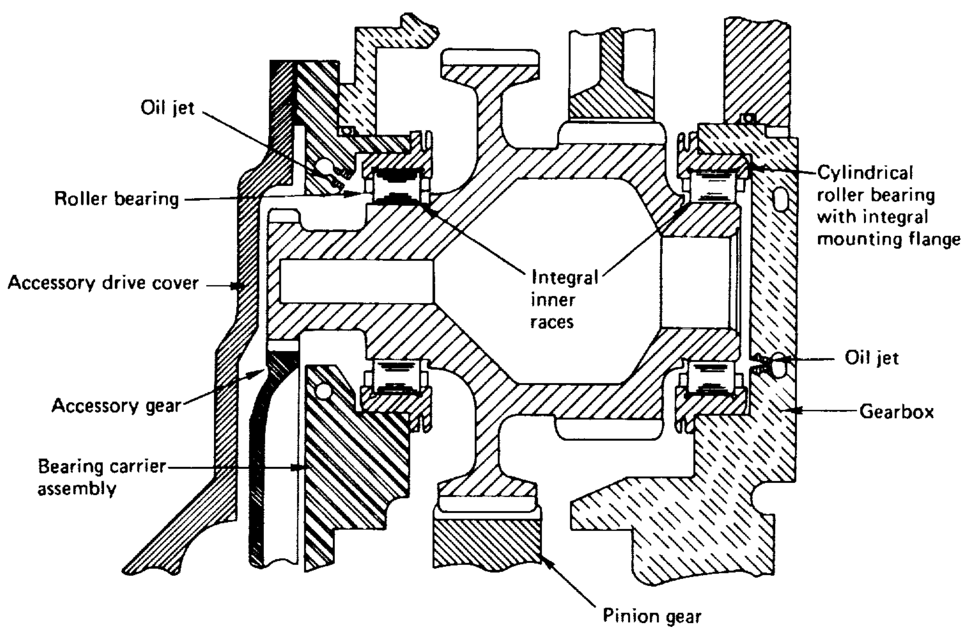
(c) Duplex DF (face-to-face)

Figure 2.—Angular-contact ball bearings.

Cylindrical roller bearings possess several times the radial capacity and stiffness of ball bearings. They are used only for radial loading; axial loads are taken by ball bearings axially connected to the roller bearing shaft (fig. 3(a)). Occasional uses of roller bearings

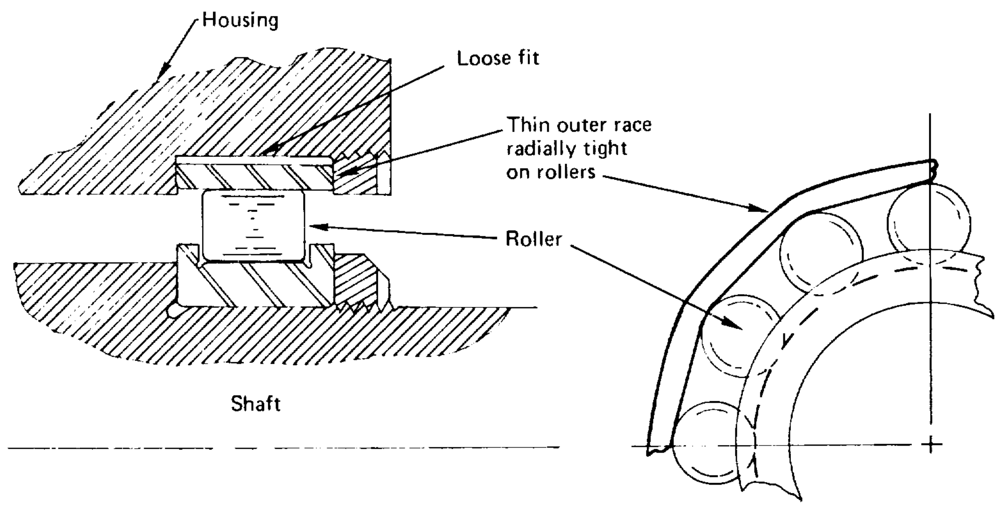


(a) Elimination of thrust load from roller bearing



(b) Cylindrical roller bearings used for axial location

Figure 3.—Roller bearing applications.



(c) Radially tight roller bearing

Figure 3.—Concluded. Roller bearing applications.

for axial loads are limited to handling the loads that result from providing axial location for rotating components (fig. 3(b)). Roller bearings with negative diametral clearance can be used for propellant-cooled service (liquid hydrogen) (fig. 3(c)) (ref. 8, pp. 505-518).

Film bearings (e.g., balance pistons) are prevented from rubbing contact during startup until sufficient pressure is generated to lift the bearing. Figure 4 shows typical balance piston and bearing configurations. Figure 5(a) is a schematic of the balance piston/bearing arrangement; while figure 5(b) is a load-deflection diagram with corresponding clearances and deflections. The values will vary with individual turbopump designs; however, the basic objective is to prevent balance piston contact from occurring (fig. 5(b), V and VI). This is accomplished by selecting gaps 3, 4, 5, and 6 in fig. 5(a), so that the loads at V and VI exceed the net pump shaft loads when insufficient fluid is pumped for balance piston operation.

When the need for adequate directional load support capability has been satisfied, the requirement that the bearing possess sufficient load capacity must then be considered. Load capacity is a direct function of bearing size unless high-speed effects predominate. For low-speed applications ($DN < 0.5 \times 10^6$), the bearing is sized by selecting a sufficiently large bearing from manufacturers' catalogs. For high-speed applications, this method is inadequate. To determine the necessary capacity for required life, the designer must resort to load-life-speed relationships that account for centrifugal, inertial, and friction forces.

The relations for dynamic analysis of ball and roller bearings accounting for speed effects are derived in reference 3. This analysis has been programmed for digital computers, making possible rapid determination of high-speed bearing stresses, life, stiffness, dynamic forces, and velocities.

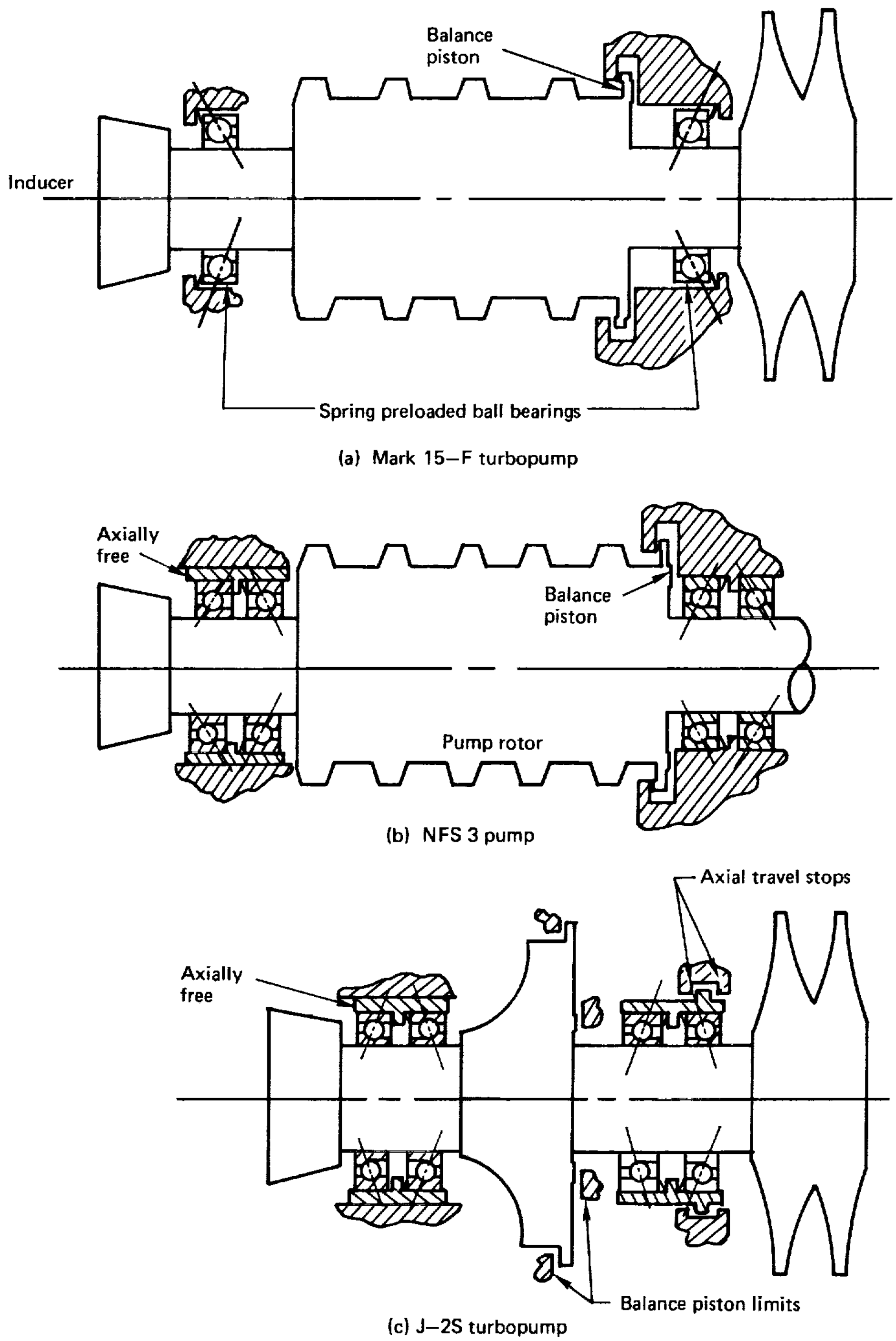
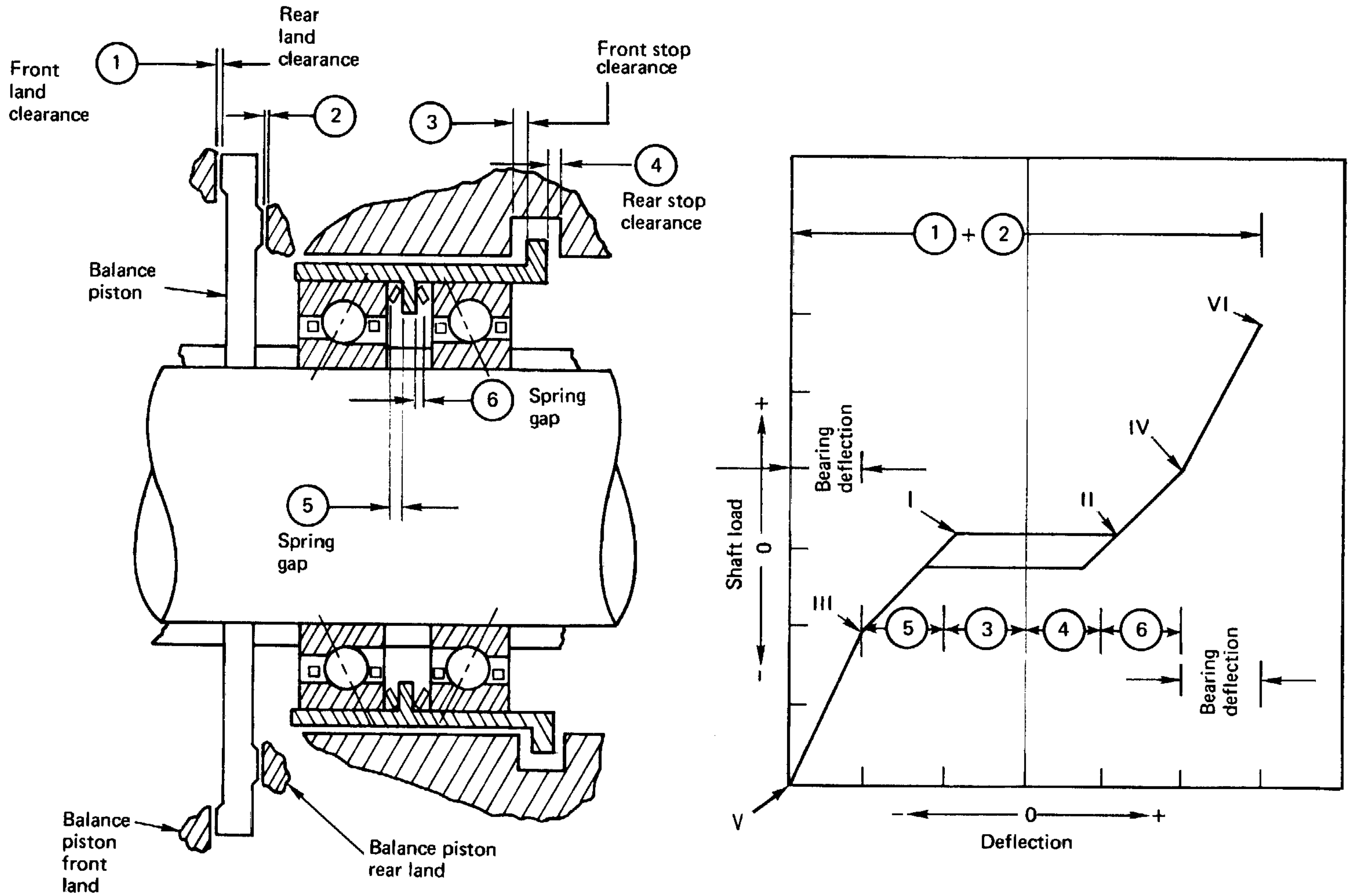


Figure 4.—Balance piston bearing systems.



(a) Balance piston schematic

(b) Load and deflection chart

Figure 5.—Balance piston system clearances, loads, and deflections.

Turbopump required service life is usually 1 to 2 hours. Bearings are sized for this service by specifying a calculated B_{10} fatigue life (including speed effects) that exceeds turbopump life by an arbitrary factor (usually 10). This conservative value is used not only to prevent fatigue failures, but also to compensate for other undefined life-reducing factors such as the low lubricity of most propellant/coolants. B_{10} fatigue life is the operating time that 90 percent of identical bearings will exceed without fatigue failure. It is calculated by relating the stress cycles of a new design to the statistically analyzed results of repeated fatigue tests on controlled test bearings. Most bearing manufacturers present basic life data on their bearings; however, their life rating methods do not include effects of centrifugal force. These effects are considered in the general analysis (ref. 3).

2.1.2 Speed Capability

DN value (product of bearing bore in millimeters and shaft speed in rpm) is used as the general measure of bearing speed because it is simple, easy to compute, and is widely understood and accepted. DN is not completely descriptive of speed severity as it does not account for rolling element size or bearing internal geometry; however, no entirely acceptable substitute for DN has been proposed.

Table III lists the approximate maximum speeds that have been achieved for bearings in operational turbopumps and experimental programs using the coolant shown. Maximum attainable speeds have not been established for each bearing type and coolant. However, it is reasonable to expect that the speed levels in table III can be exceeded provided that the propellant coolant has a higher rank as a lubricant than the given coolant (sec. 2.1.7).

TABLE III.—Speed Limits for Bearings

Bearing type	Existing maximum turbopump DN, million	Existing maximum test DN, million	Limiting factor	Test coolant
Conrad-type ball	1.6	1.6	Cage weakness	Liquid hydrogen
Angular-contact ball	2.05	3.0 (ref. 4)	Heat generation	Liquid hydrogen
Cylindrical roller	1.6	1.6	Roller guidance, cage slippage	RP-1

In turbopump designs, bearing speeds near the ultimate bearing speed capability are avoided because operation near this limit makes all aspects of alignment, loading, cooling, lubrication, and manufacturing variables critical to success; the slightest departure from ideal conditions may result in failure. Therefore, to gain reliability, DN values are held to the lowest possible levels consistent with pump requirements.

It should be noted also that a finite minimum load (inherent application loads or preloads) is required at each bearing speed to prevent gross skidding of rolling elements and consequent damage to bearing surfaces. The minimum required load is a function of materials, bearing geometry, lubrication, and acceptable life reduction. Dynamic analyses of bearing performance (ref. 3) are required to determine the skidding tendencies of an application. A discussion of roller bearing skidding appears in reference 1, pp. 321-328.

The sensitivity of bearings to speed, load, and life changes (as are often required in uprating) is illustrated in the two examples below:

(1) Conrad bearings with phenolic cages, entirely satisfactory for turbine shaft thrust bearing service at 1.2×10^6 DN in Atlas and Thor vehicle turbopumps, had to be replaced by split-inner-ring bearings with one-piece bronze cages when the speed and load were increased by approximately 10 percent for service in the H-1 engine.
(2) Extended service life requirements in J-2 engine fuel pump bearings operating at 1.6×10^6 DN caused failures of two-piece, riveted cages made of Armalon (glass-fabric-supported polytetrafluoroethylene (PTFE)). The substitution of a one-piece (otherwise identical) cage resulted in satisfactory service.

2.1.3 Stiffness

Bearing stiffness is a controlling factor in shaft dynamics (critical speed control). Therefore, bearing detail design calculations are often made and used as background information for decision making during selection of the shaft layout. Dynamic analyses (ref. 3) programmed for digital computers are used to determine quickly the radial stiffness of candidate bearing designs for the expected range of loads and speeds. Bearing type, preloading, mounting, and shaft layout selections are often determined on the basis of calculated bearing spring rates (stiffness). Roller bearings are sometimes used to obtain adequate stiffness (1.5 to 2.0×10^6 lb/in.) when subcritical operation is required, e.g., in throttled engines. Supercritical operation can be obtained by the use of soft mountings for bearings, thus depressing criticals to below operating speeds.

On one particular experimental turbopump a radial stiffness of 120,000 lb/in. was required for critical speed control. Analysis indicated that at the desired lower preload limit of 40 lb, the radial stiffness had dropped off to 20,000 lb/in.; the solution required increasing the preload to 60 lb minimum even though this reduced the calculated fatigue life of the bearing. This type of compromise often is required in a high-speed, relatively lightly loaded ball bearing because the centrifugal force per ball is large relative to the total ball loads.

2.1.4 Misalignment Tolerance

Misalignment tolerance of ball bearings can be increased by providing sufficient cage pocket clearance (ref. 5) to allow for circumferential advance and retard of ball position caused by ball speed variation (BSV). Excessive BSV results in cage pocket wear and heating. In reference 6, the detrimental effects of BSV are discussed and methods are provided for calculating BSV. Some designers prefer to use elliptical ball pockets. With this technique, only the circumferential cage pocket clearance is increased while the standard axial cage pocket clearance is unchanged; this practice prevents cage wobble and consequent land wear. Digital computer programs based on the original work presented in reference 3 are used to compute cage-rolling element forces and rolling element excursions that determine the required minimum clearance.

Roller bearing misalignment results in capacity loss caused by edge loading of the rollers. Misalignment tolerance is increased by crowning the outer race (fig. 6) (ref. 7). In general, the misalignment imposed on turbopump bearings is minimized by proper shaft and housing design and manufacturing techniques such as line-boring of gear cases.

2.1.5 Bore

Turbopump bearing bores have ranged from 8 to 170 millimeters, and testing has been conducted on bearings with 200 millimeter bore. The preferred bore size is the smallest diameter that satisfies the shaft strength, stiffness, and bearing capacity requirements. For extremely high-speed applications, an increase in bore size will not always result in an increase in bearing load capacity because centrifugal loading of the rolling elements increases faster than capacity as size increases. In such cases, tandem duplex thrust bearings are utilized (e.g., in the turbopumps for the F-1 and M-1 engines).

Placing splines inside bearings is avoided because the bore must expand to accommodate two torque-carrying shaft sections. In addition, uneven expansion of torque-carrying splines tends to distort bearing races into cone shapes; this results in roller skewing and consequent end wear.

2.1.6 Internal Clearance

Unfitted diametral clearances are specified with allowances made for all causes of diametral clearance loss. Except as noted below, care is taken to prevent the occurrence of negative clearance, which almost always results in sudden catastrophic failure. Press fits, centrifugal growth, material properties (such as Poisson's ratio), and thermally induced differential growths are all considered in establishing design clearances.

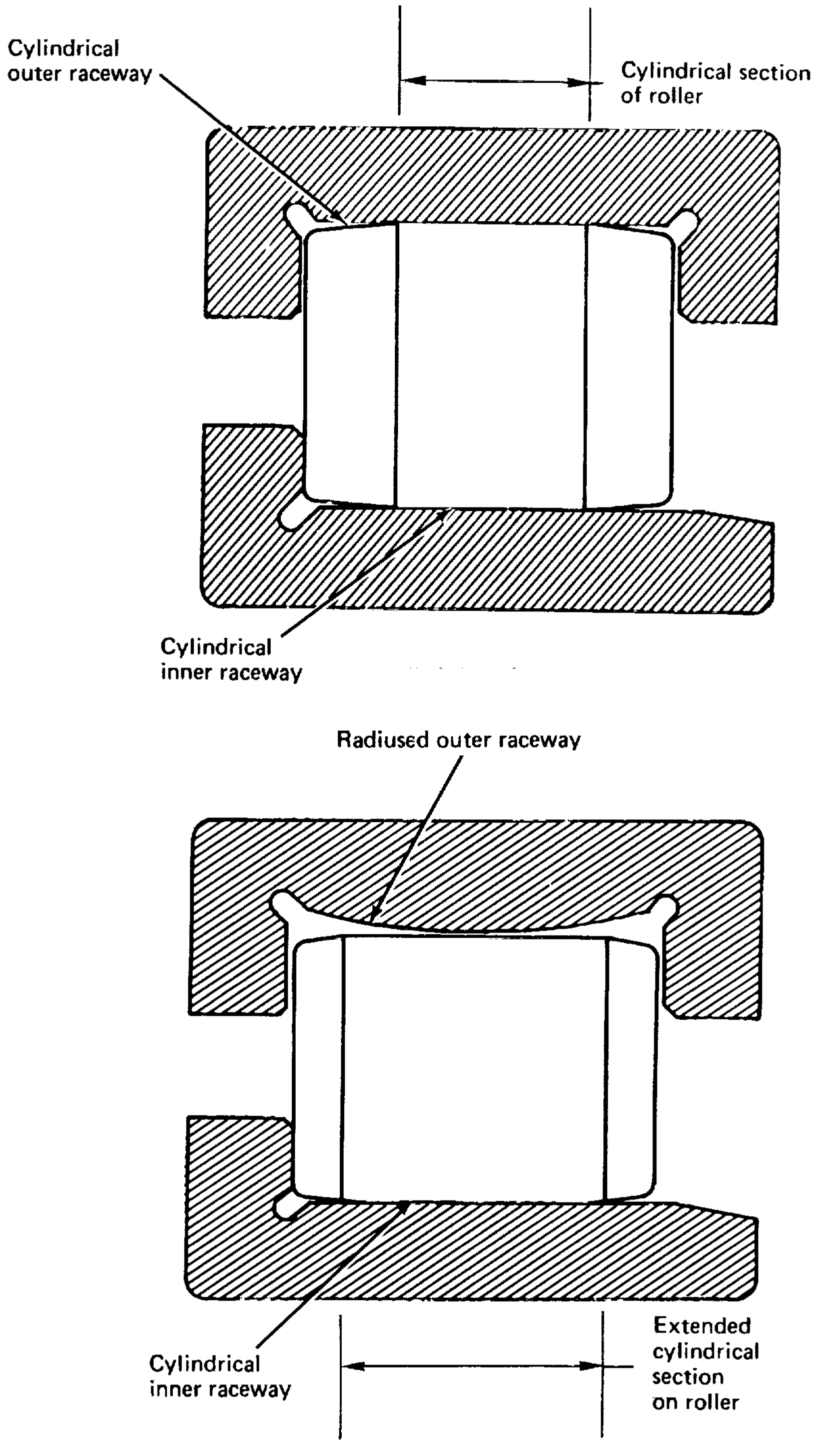


Figure 6.—Race crowning to allow for misalignment in roller bearings.

Roller bearings are sometimes designed utilizing radial preloading where light-load, high-speed conditions will result in excessive roller skidding and wear. Preloading techniques include elastically deforming the outer races out of round (ref. 1, p. 323) and use of zero or negative diametral clearance. Special design considerations for radially tight bearings are given in the unclassified appendix of reference 8, vol. III, pp. 505-518. In utilizing zero or negative clearance, extreme care must be taken to ensure that sufficient cooling is available to prevent a self-aggravated thermal growth loading cycle that will rapidly result in a complete bearing seizure.

2.1.7 Cooling

Table II summarizes current bearing cooling practice. Most turbopump bearings are cooled with pumped propellant. A small quantity of the total pump flow is bled from a high-pressure region of the pump, flows through the bearings, and re-enters into a low-pressure area of the pump or at the pump inlet. An investigation of the bearing lubricating ability of various liquid rocket propellants (ref. 18) resulted in the following ranking (in descending order): RP-1, liquid oxygen, liquid hydrogen, N_2O_4, IRFNA, EDA, UDMH, N_2H_4, and 50-50 N_2H_4-UDMH.

A few turbopumps still use conventional lubrication that requires a separate tanking and circulation system. Oil systems in use include recirculating (Titan) and once-through overboard-drain (Atlas and Thor).

Coolant effects.—Roller end wear of cylindrical roller bearings is aggravated when roller bearings are used with chemically reducing coolants such as liquid hydrogen and the amines (50-50 mixture of UDMH and hydrazine). Roller end wear does not appear to be severe with liquid oxygen and is insignificant with N_2O_4 (refs. 2, 9, and 10). The use of dissimilar roller and race materials will reduce the end wear significantly even with chemically reducing coolants (ref. 9).

For high-speed turbopumps, angular-contact ball bearings with one-piece, outer-land-riding cages made from self-lubricating materials containing PTFE are preferred. These bearings have been tested successfully at speeds to 3.0 x 10^6 DN (ref. 4) (table II).

Development of roller bearings for propellant-cooled high-speed use requires special considerations in clearances, guidance, and roller skidding prevention (refs. 2, 8, 9, and 10).

Coolant quantity.—Coolant quantity is established by heat balance calculations confirmed or modified by testing. Typical values for coolant flow range from 0.1 gpm of RP-1 for low-load, low-speed bearings to 150 gpm of liquid hydrogen for large, high-speed bearings (table II). With roller bearings cooled by liquid hydrogen, increasing coolant flow above that required for heat balance has reduced roller end wear.

Coolant delivery system.—Jets of lubricant carefully targeted to the gap between the inner race and cage are used for the more viscous conventional oils and most RP-1 systems (fig. 3(b)). The effectiveness of this lubrication system depends on the accuracy of the jet targeting and the elimination of other possible reasons for failure of adequate lubricant to reach all necessary areas of the bearing. Churning losses are less than those with flooded lubrication systems, decreasing the flowrate required (especially with viscous fluids where $\mu > 1$ centistoke) and thereby decreasing the payload penalty of the lubrication system. Additional coolant flow is sometimes directed through slots in the bearing races or shaft under the races. Most lubrication systems for highly loaded power transmissions used at low ambient pressures (altitudes of 100,000 ft or more) are pressurized to 2 psia to provide lubricant circulation by windage. Otherwise, lubricants may not reach critical areas of the bearing and the bearing will fail.

Tests with the Atlas and Thor engine transmissions have demonstrated that outer-land cage-equipped ball and roller bearings with direct lubrication jet cooling can operate successfully at very low ambient pressure (2 mm Hg). The Agena engine employs an unpressurized splash-lubricated gear transmission (no lubrication jets) that has operated successfully at 0.2 psia.

Flooded through-flow systems are generally used for propellant-cooled bearings for two reasons:

(1) There is less chance of blockage of jets with particles of foreign matter, or with pieces of ice in cryogenic systems.

(2) Churning losses and heating are small because of the low viscosity of most propellants.

Coolant quality (liquid/vapor fraction).—Attempts are made to maintain the coolant at 100-percent liquid condition at the bearings because this increases the cooling effectiveness. When the fluid temperature in the pump is above the critical temperature, the pressure is sometimes altered to be above the critical pressure (ref. 11). Gaseous hydrogen has been used in the RL-10 engine turbopump as a bearing coolant; flowrate was found to be critical to successful bearing operation.

Contamination.—Chemical contamination of propellant systems is controlled by applying cleanliness standards similar to MSFC-SPEC-164 (developed for liquid-oxygen systems) (ref. 12), which controls particle size, particle number, and nonvolatile residue. In addition, passivation by controlled exposure to fluorine gas is required for components intended for service in liquid fluorine or in FLOX (mixture of liquid fluorine and liquid oxygen).

When bearings are not corrosion resistant, water is excluded from the bearing. In oil and RP-1 systems, this is accomplished by use of water-displacing preservatives; in cryogenic systems, by use of dry gas purges (MIL-P-27401B nitrogen (ref. 13) or MIL-P-27407 helium (ref. 14)) or desiccants (MIL-D-3464D (ref. 15)).

Filters and screens are used to prevent blockage of lubricant jets and damage to bearings by particulate contamination. In RP-1 and oil systems, 40-micron filters are used, while screens with 10-micron openings are used in the flooded coolant system in the J-2 engine liquid-oxygen turbopump. The J-2 engine fuel pump uses no filter or screen, but relies on centrifugal force to separate particles from the coolant and prevent contamination of the bearings.

The RL-10 engine turbopump depends upon the initial cleanliness of components and system. In the Titan II, turbine exhaust gases used to pressurize the gear box generate sludge in the lubricant (MIL-L-7808 (ref. 16)); replaceable filters are used in the turbopumps to remove this sludge. A bypass check valve is used to prevent lubricant starvation if the filter clogs.

2.1.8 Bearing Mounting

Interference fits.—A light press fit between the rotating race and its mounting is maintained to prevent relative motion and consequent fretting. For heavy radial loads, heavier press fits are required. Some applications (H-1 engine turbopump; Thor and Atlas vehicle turbopumps) experienced extremely heavy radial loads, and creeping of races could not be prevented by clamping. The problem was solved by making the inner races integral with the gears (figs. 2(a) and 3(b)).

Maximum interference fits are held below the fit that causes inner race fracture. Interference fits up to 0.001 in./in. of bore diameter are used without problems.

Sliding fits.—Spring-preload bearings and axially floating bearings as used in the turbopumps for the J-2 engine require loose-fit outer races to maintain axial freedom. When sticking of the outer races, coupled with shortening of the rotor due to the Poisson effect of centrifugal force, caused loss of preload and shaft dynamic problems in the engine fuel pump, looser fits for the bearings and increased preload were used.

Bearing mounting surfaces.—Shaft journals and housing bores generally are chrome plated to prevent galling and fretting of the surfaces due to the relative motion of the races or assembly press fitting. This practice is mandatory for Inconel and Monel shafts used in cryogenic pumps. Titanium shafts cannot be chrome plated, but the contraction is similar to 440-C steel; therefore, lighter fits are possible, and galling because of assembly and disassembly is minimal. SAE 4130 steel shaft journals hardened to 40 R_c generally do not gall even with heavy press fits.

Corner radius interference.—Interference of the bearing bore and outer diameter corners with the shaft shoulder corner radius is prevented by use of a grind relief (fig. 3(a)).

Mounting surface accuracy.—The mounting surface accuracy (roundness of cylindrical surfaces, taper of diameters, and axial runouts of shoulders) is maintained by specifying tolerances equivalent to ABEC 5 or RBEC 5 (AFBMA Standards (ref. 17)). Looser tolerances would destroy the accuracy obtained from the bearing design.

Shoulder heights.—Shaft and housing shoulder heights are chosen to provide sufficient compressive strength to transmit clamping loads through the solid race sections without bending the race and possibly pinching the rollers. Normally, 0.050 in. or more of the bearing race face is exposed to allow pulling of the bearing. When this is not possible, bearings are provided with grooves or they are disassembled by pulling through the rolling elements, a practice that often damages the bearing.

Race retention.—Rotating races are retained by nut clamping (fig. 7(a)) or tension bolting (fig. 7(b)). Early designs that relied on press fit alone were altered when race walking caused fretting of the journals and axial displacement of the races.

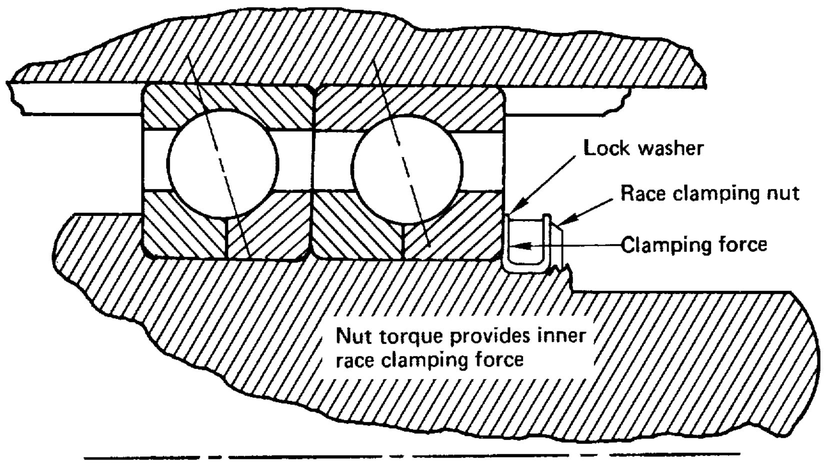
Snap rings for turbopump bearing race retention have been abandoned because of the possibility of removal by centrifugal force, shock, and vibration.

When either of the bearing races rotates relative to a heavy radial load, the clamping load must be large enough to prevent race creep. Figure 8 illustrates the Poisson effect on the inner race of a heavy radial load. This rotating "bulge" of metal caused severe fretting of Atlas turbopump gear bearing inner races. Extremely strong retainers did not stop the fretting. The final solution was to make the inner races integral with the gear (figs. 3(a), (b)).

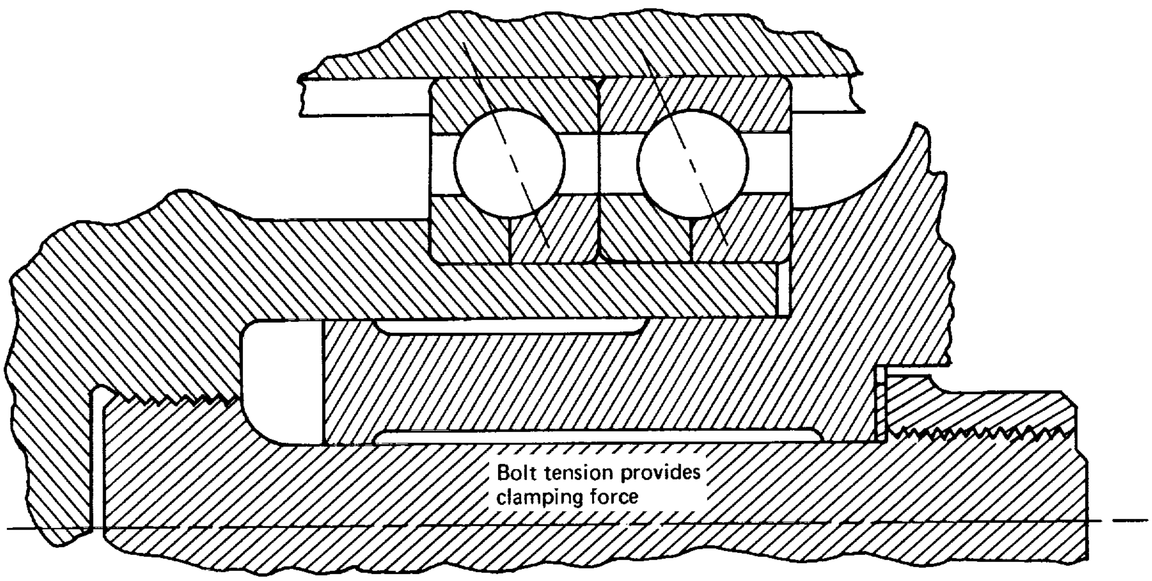
The bearing-retaining nut or tension bolt is designed and applied to provide clamping loads large enough to prevent the stack of components from loosening under thrust, bending, or thermal effects. Clamping loads as large as 50,000 pounds are used in the bearing tension bolt in the J-2S engine. A clamping-nut torque of 800 ft-lb is used in the F-1 engine turbopump. A case of severe fretting of the gear and bearing race mating surfaces (fig. 9) in the Atlas sustainer turbopump was eliminated entirely by increasing the clamping-nut torque to 200 ft-lb.

2.1.9 Bearing Materials

Corrosion resistance.—Bearings are made of corrosion resistant materials for propellant-cooled applications. Races and rolling elements are made of 440-C steel; cages are made of plastics, and occasionally are reinforced with corrosion resistant steel (CRES) or aluminum. Standard noncorrosion-resistant materials (52100, tool series) are protected during idle periods by preservatives. Grade 440-C steel, the most com-



(a) Nut clamping



(b) Bolted retention

Figure 7.—Race-retention methods.

monly used race material for propellant-cooled applications, is not entirely corrosion resistant, and investigations are underway to discover a substitute; none has been found with the combination of hardness, corrosion resistance, fatigue life, and availability displayed by 440-C. Nickel-based alloys are corrosion resistant, but soft. Sintered carbides are corrosion resistant and hard, but brittle. Stellite Star-J has been investigated as a cryogenic bearing material, but testing has not produced design recommendations.

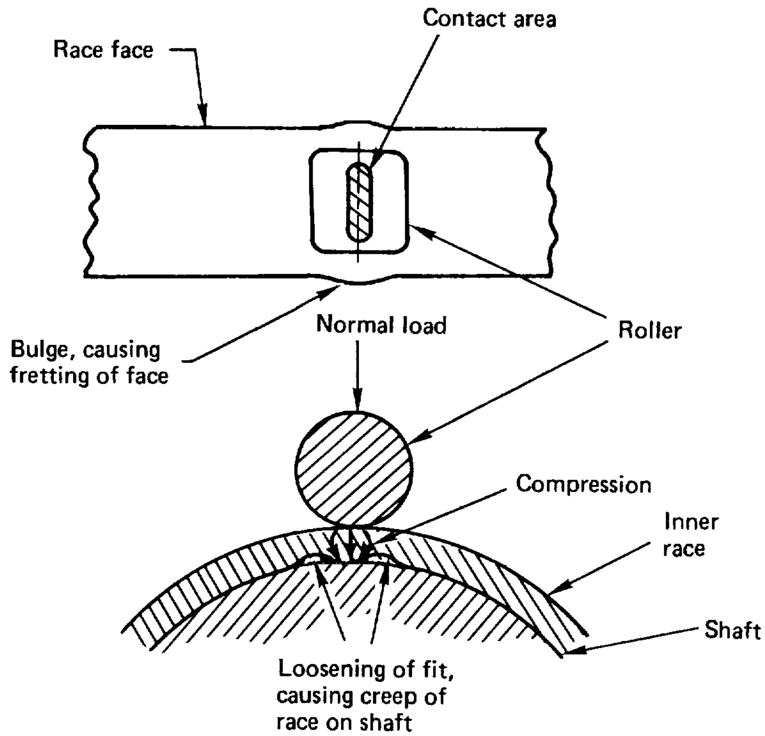


Figure 8.—Poisson's effect of heavy radial load.

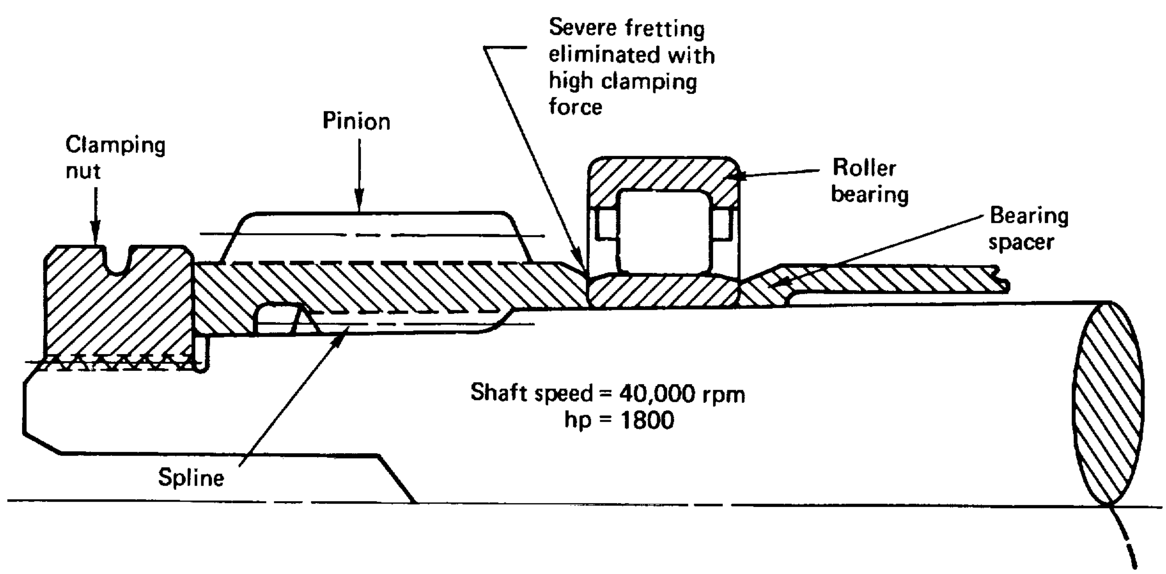


Figure 9.—Bearing clamping to eliminate fretting.

Hardness.—Since full capaciy requires a minimum hardness of 58 R_c, few materials qualify as full capacity race and rolling element materials. In addition to the materials that appear in table II, M-10 and M-50 tool steel have been used. To date, materials softer than 58 R_c have not been used in turbopump bearings. Extensive testing would be required to provide reasonable confidence in the life capabilities of softer materials.

Adhesion resistance.—Adequate resistance to adhesion between the rolling element and race is attained for most propellant environments using 440-C (table III). Oil and RP-1 pose no problems for 52100 or tool steels. However, 440-C displayed adhesive degradation of races and balls when cooled by strong reducers (N_2H_4, UDMH, and mixtures of both) (refs. 18 and 19). In reference 9, however, successful testing of the same material combinations is reported. Additional testing is required in this area of conflicting results. As noted earlier the use of dissimilar race and rolling element materials reduces wear rates in chemically reducing coolant environments (ref. 9).

Dimensional stability.—Stabilization by repeated cycles of chilling with liquid nitrogen and then tempering has successfully eliminated dimensional changes noted in liquid-oxygen and liquid-hydrogen bearings. Stabilizing chill cycles are part of most cryogenic bearing design specifications for 440-C bearings. Requirements for stabilization procedures for nickel-based alloys and carbides are not known.

Grinding burns.—Grinding burns once were a source of dimensional changes and internal stresses that resulted in race fracture. These have been largely eliminated by more rigid controls of grinding-wheel grit speed and feed and of coolant type and amount. Nital etch will detect grinding burns in 52100 and 4620 steels, but no satisfactory test has been proposed for grinding burn detection for 440-C steel. (Picric etch has been proposed as a detection method, but its effectiveness has not been proven.) Grinding burns are not a significant problem area if effective quality control methods are used to ensure proper processing.

Material cleanliness.—Vacuum melted steels are used extensively for race and rolling elements to ensure freedom from nonmetallic inclusions and consequent increase in fatigue life (refs. 20 and 21). Consumable electrode vacuum melting (CEVM) is preferred, as it produces more consistently clean steel than can be obtained by use of other vacuum melting processes. The reduction in nonmetallic inclusions obtained by vacuum melting also reduces the possibility of chemical attack by reactive propellants.

Cage material friction.—Low-friction materials are used for cages; those used for propellant lubrication provide their own lubricity and in addition may lubricate the rolling surfaces by transfer of a low-friction film from cage to balls to raceway (ref. 22). Table II presents the cage materials used with the various propellant/coolants. An exception in the self-lubricating quality sought for propellant-cooled service is the K-Monel and other high-nickel alloys used with liquid fluorine. It is possible that low-shear-strength nickel fluorides form on the cage and provide a slick film on the surface. Testing is continuing on bearings cooled by liquid fluorine.

2.1.10 Testing

Bearing designs are subjected to full-scale empirical tests in test rigs or in turbopumps to evaluate coolant flow and geometric configurations including tolerance range. Great care is taken to assure that test conditions simulate as closely as possible the expected operating condition. Bearings often are made separable to permit periodic inspection of bearing components during test programs.

Subscale tests of rolling element, race, and cage materials are often used for screening candidate new materials. The value of these tests is limited by the accuracy of simulation of the actual bearing operating conditions; therefore, the results of simple rolling and sliding tests must be interpreted with care and confirmed with full-scale tests.

Rolling contact tests.—Subscale rolling contact screening tests are used to select bearing race and rolling element materials for later fabrication of full-scale bearings. An example of this procedure is illustrated in reference 19 where a Barwell conversion of the Shell four-ball machine was used to screen various rolling contact material combinations.

Sliding friction tests.—Simple button riders held in contact with a spinning plate have been used extensively in cage material screening tests (refs. 18 and 19). The information from the testing described in reference 23 was used directly in selecting successful cage materials for liquid-hydrogen and liquid-oxygen service for the J-2 engine turbopumps.

2.2 Bearing Component Design

2.2.1 Rolling Element Design

Ball and roller element complements (size and number) are selected by performing dynamic analyses of candidate bearing designs (ref. 3), with the following considerations:

- Maximum ball or roller size is limited to the diameter whose further increase results in a decrease in capacity because of centrifugal force effects (contact stress, spinning and sliding). Ball sizes for turbopump bearings range from about 3/16 to 7/8 in. diameter; roller sizes range from approximately 1/4 to 1 in. diameter. Where capacity requirements exceed the increase available by increasing diameter, tandem duplexed bearings are used. Minimum ball or roller size is limited to the diameter that results in an adequately strong cage cross section and adequate coolant flow. Cage radial thickness of 0.100 in. and coolant flow radial space of 0.070 in. are generally the minimum used for bearings of over 40-mm bore.

- The maximum number of rolling elements is limited by cage web thickness considerations. It was found (ref. 9) that for nonmetallic, propellant-cooled bearing cages, a minimum cage web thickness of 0.150 in. is desirable. Metallic cages can utilize narrower bars because of greater strength. In addition, metallic cages are not adaptable to sacrificial wear as are low-friction, nonmetallic cages because metallic wear debris is potentially more detrimental to the bearing.
- The minimum number of rolling elements is not a usual design goal. The typical minimum number occurs in a Conrad-type bearing; the space between balls is slightly less than ball diameter.

Ball and roller diameter uniformity.—Ball and roller size variation is limited in new bearings to 10 to 20 microinches (μ in.). Excessive size variation leads to orbital velocity variation that results in cage distress. Ball quality criteria have been standardized by industry into AFBMA standard grades. These grades specify diameter uniformity among individual balls (in a bulk container) and ball roundness in an individual ball. Balls for turbopump bearings are usually AFBMA grade 10 (AFBMA Standards (ref. 24)).

Ball and roller roundness.—Ball and roller roundness is specified to limits of 10 to 20 μ in. to ensure smooth running bearings and load sharing among rolling elements. Excessive size variation will result in heavy cage wear and potential fatigue failure.

Ball and roller surface finish.—AFBMA Standards are utilized to specify ball quality; in addition to roundness and size variation, these standards also control ball surface finish. The AFBMA grade 10 specified for turbopump bearing balls controls the surface finish to 1.0 μ in. arithmetic average (AA) maximum.

Presently, a roller finish must be specified separately as there is no AFBMA standard for rollers. Typical roller surface finishes are 4 μ in. AA on outer diameters and 6 μ in. AA on the roller ends.

Roller guidance.—Roller guidance is extremely important to satisfactory wear performance of rollers and is ensured by control of the following design features:

(1) Roller length/diameter ratio of 1 is preferred; 1.3 is used as a maximum.
(2) Roller ends are held square within 0.0001 in. of the total indicated runout.
(3) Roller bearing race shoulders are 14 to 20 percent of the roller diameter, providing a guiding contact chord of 75 percent of the roller diameter.
(4) Roller end clearance between guiding lips in the guiding raceway may range from 0.0002 to 0.0020 in. in turbopump bearings. (Care is taken to avoid roller pinch from clamping loads.) Roller end clearance uniformity within an individual bearing assembly is assured by specifying roller length to be uniform within 0.0002 in.
(5) Roller corner radii concentricity control of 0.002 in. TIR (total indicated runout) is necessary to avoid roller cage wear caused by roller unbalance. Bearing manufacturers are now developing techniques that will enable them to supply rollers with the end radii concentric within 0.0005 in. TIR.

Roller corner radii.—Roller corner radii are controlled so that the maximum value still results in adequate roller guidance, but the minimum does not interfere with race shoulder corner radii. Typical turbopump bearing roller corner radii are controlled to be within the range 0.020 to 0.030 in.

Roller end relief (crowning).—To prevent stress concentration at the roller end, rollers are crowned (fig. 10) an amount slightly greater than the elastic deflection of the heaviest loaded roller. In addition to the amount for end relief, crowning provides an allowance for misalignment.

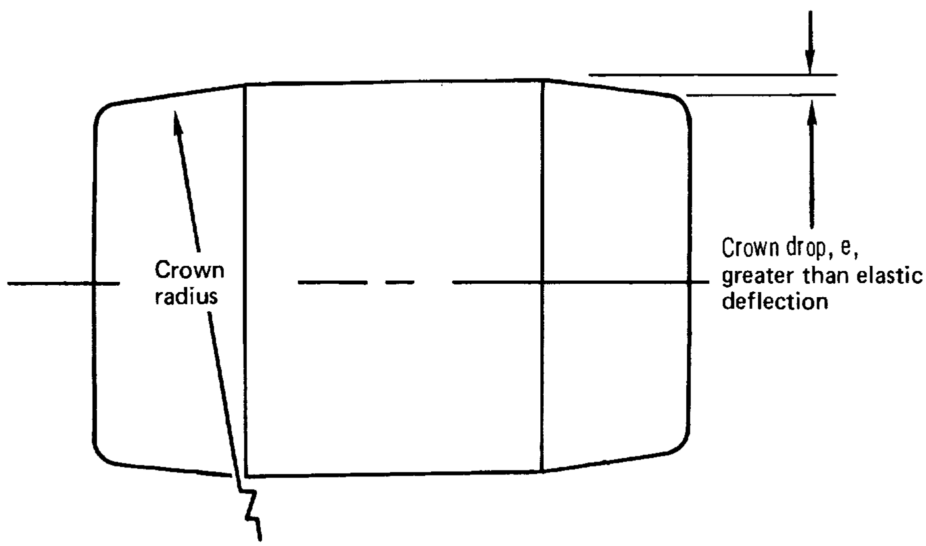


Figure 10.—Roller crowning.

2.2.2 Race Design

Race cross section.—A race cross section thickness less than 0.04 times the mean diameter promotes brittle fracture of 440-C races used in liquid-hydrogen service, and therefore greater thicknesses are used.

Ball bearing transverse race curvature.—Transverse race radii less than 51 percent of the ball diameter result in such wide contacts that excessive spinning and sliding occur. The breakover point where closer curvature results in excessive heat generation rate varies with speed and bearing size. In a bearing with close curvatures, small changes in clearance make large proportional changes in contact angle. The bearing geometry is quite sensitive to manufacturing tolerances, and dimension variations that would be insignificant in a bearing with 54 percent curvatures would make rapid changes in diametral clearance and contact angle in a bearing with 51 percent curvatures.

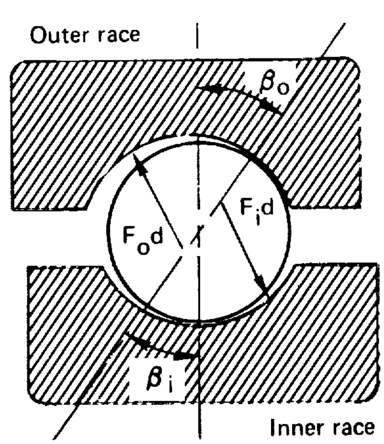
To reduce heat generation in high-speed ball bearings, race curvatures are made wider than the normal 51 to 53 percent. The maximum race curvature presently used in turbopumps (J-2 engine fuel pump) is 59 percent. A further increase in curvature increases stress and decreases capacity, with diminishing returns in spin reduction.

Inner (F_i) and outer (F_o) race curvatures are varied independently to achieve specific design goals as indicated in table IV. Most rules and guidelines used are the

TABLE IV.—Curvature and Contact Angle for Specific Design Goals Using Ball Bearings

Goal	Bearing design parameter		
	*F_o	F_i	β, degrees
Maximum thrust capacity	Minimum ~0.52	Minimum ~0.52	Maximum ~35
Maximum radial capacity	Minimum ~0.52	Minimum 0.52	Minimum ~12
Minimum contact angle divergence ($\beta_i - \beta_o$)	Maximum ~0.59	Maximum ~0.59	Minimum ~12 to 15
Minimum torque	Maximum ~0.59	Maximum ~0.59	Minimum 12 to 15
Maximum radial stiffness	Minimum 0.52	Minimum 0.52	Minimum 12 to 15
Inner race control	Maximum >0.54	Minimum <0.54	Minimum 12 to 15

*F = race radius divided by ball diameter.



result of analysis, with little if any empirical confirmation in full-scale bearings. In general, if a combination performs satisfactorily during testing, the design is adopted and no further experiments are performed, partly because testing, especially with cryogenic coolants, is extremely expensive. Some results of rig testing at extremely high speed (ref. 4) have indicated that designing for minimum heat generation allows operation at 3 x 10^6 DN with large (150 to 200 mm) bearings. In general, bearings for liquid-hydrogen service are required to operate at high speed because of the low density of hydrogen. These bearings are therefore designed for a low-heat-generation maximum-speed capability.

Optimum high-speed bearing design seeks to minimize the detrimental inertial effects of speed. This requires minimizing initial contact angle β to reduce the dynamic contact angle divergence $(\beta_i - \beta_o)$. Large total curvatures $(F_o + F_i - 1)$ permit lower contact angle while maintaining a required safe diametral clearance (see sec. 2.1.6).

Within the total curvature required for adequate clearance, inner and outer race curvature can be varied, and the designer is free to select a beneficial combination F_o/F_i. There is theoretical justification for attempting to obtain inner race control (IRC) (refs. 25 and 26) by setting $F_i < F_o$. IRC exists when rolling action is maintained at the inner race contact and spinning occurs predominantly at the outer race contact. The advantage of IRC is that it produces less bearing torque than ORC (outer race control) and allows small F_i, resulting in lower contact stress and longer inner race fatigue life. The limitation to the benefits obtained in this way are that (1) it is difficult to obtain IRC in bearings of over 50 mm bore, and (2) at extremely high speeds, the centrifugal force acting on the balls makes the outer race stress the more significant life-controlling factor. In general, in extremely high-speed bearings F_o is made less than F_i. This reduces heat generation caused by ball spinning at the inner race contact without unduly affecting bearing fatigue life, which is limited primarily by the outer race because of centrifugal ball forces.

<u>Race finish</u>.—Although extremely fine finishes are possible, the following finishes are easily attainable and give satisfactory service:

Direction	Surface[1] roughness	Waviness[1,2]	
		17 to 330 peaks/revolution	4 to 17 peaks/revolution
Raceway, circumferential	6 μ in. AA	30 μ in.	60 μ in./30° circumferential arc
Raceway, transverse	30 μ in. AA	NA	NA
Cage land, circumferential	25 μ in. AA	75 μ in.	NA

[1]See ASA B46.1 - 1962 (ref. 27) for surface texture definitions.

[2]Waviness is normally defined as the maximum peak-to-valley dimension; however, waviness of bearing races is often defined as the average deviation from a nominal surface (see ref. 27).

Some users require even finer finishes in special cases. Bearing manufacturers can obtain a transverse finish of 6μ in. AA with resulting improvement in circumferential finish. Production techniques for providing these finer finishes are under development.

Surface finish refinement promotes formation of elastohydrodynamic (EHD) films, provides smooth bearing operation, increases fatigue life, and aids in the formation of transfer films (ref. 22). Actual formation of EHD films is unlikely in turbopump bearings regardless of the raceway finish because the viscosity of the propellant/coolants is low. (For a review of current EHD technology, see ref. 28, pp. 36-37.) Although fatigue life generally increases with finer finishes, some preliminary results of single-ball fatigue testing indicate there may be a decrease in fatigue life when race finishes are reduced below 5μ in. AA.

Ball bearing shoulder configuration.—Ball bearing shoulder heights are specified as a percent of ball diameter, the value being derived from analyses that describe the dynamic position of the ball and the size of the contact ellipse. The shoulder height is specified high enough to contain the contact ellipse under all anticipated loading conditions. Normal ball bearing shoulder heights vary from 18 to 22 percent. High-speed ball bearings with low contact angles often have much lower shoulders, some only 12 percent.

To provide a safety factor for overloads, the race edge is relieved to prevent ball creasing. The value of edge relief was shown in tests of highly loaded thrust bearings for the F-1 engine turbopump; ball bearings with edge relief survived axial loads that resulted in approximately 0.030 in. excursion of theoretical contact ellipse length beyond the race edge. Bearings with no relief or inferior edge relief failed immediately.

Race relief.—Cage problems caused by ball size variation in bearings under constant thrust load are not solved simply by enlarging ball pocket size. Rather than alternately advancing and retarding from a nominal position, different size balls tend to continue to advance or retard, applying excessive forces on the cage. In limited testing at 2.0 x 10^6 DN in liquid hydrogen, a small raceway relief or unloading chute (fig. 11) 0.003 in. deep by 0.300 in. long was effective in reducing the cage forces.

2.2.3 Cage Design

Cage strength.—Cages made of weak or brittle materials are often stiffened and strengthened with metal shrouds or reinforcing bands. Cages of serviceable strength have thus been constructed from otherwise unsatisfactory materials such as filled PTFE compositions (ref. 9). The early Armalon (glass-fabric-supported PTFE) cages were reinforced with aluminum side bands, but subsequent testing and practice have shown Armalon to be rigid and strong enough at cryogenic temperatures to be satisfactory even when unsupported.

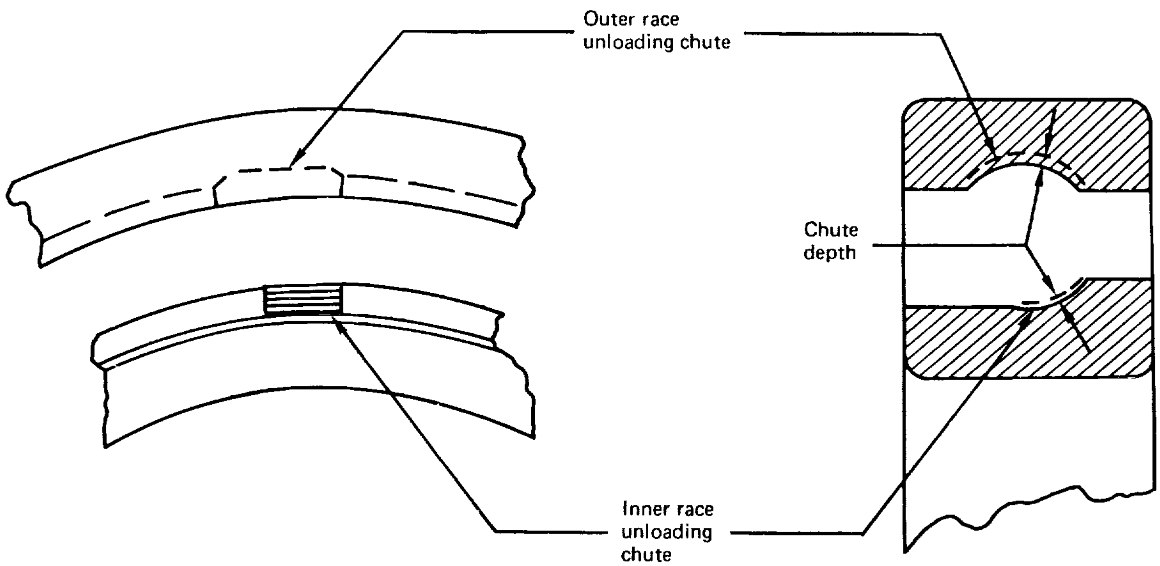


Figure 11.—Ball bearing unloading chute.

Coolant flow area.—Cages are designed with minimum cross section to allow free through-flow of coolant. Minimum radial thickness of Armalon cages used for liquid-hydrogen and liquid-oxygen bearings has been 0.100 in. for cages approximately 3.5 in. in mean diameter and 0.150 in. for cages approximately 8 in. in mean diameter. Cages of weaker materials are made thicker or are reinforced; metallic cages are usually no thinner than Armalon cages in order to preserve a generous contact area for the rolling element in the ball pockets or to avoid off-center contact with rolling elements.

Cage–rolling element contact.—The cage pocket contact with the ball or roller is usually centered at the rolling-element complement pitch line to avoid jamming or deflecting the cage as the rolling element tends to "climb" over the cage web. This is not mandatory, because in some applications the cage makes contact above the pitch line with satisfactory results. Limitations generally are not explored, and changes are made only if difficulty is encountered.

Web thickness.—Cage web thicknesses of less than 0.150 in. are unsatisfactory for propellant-cooled bearings. Metallic cage webs can be made thinner, but minimum thicknesses must be determined by testing because of differences in operating severity.

Guiding land clearance.—Cage guiding land clearance is maintained at some positive value for all operating conditions, but generally has a minimum value of approximately 0.010 in. Larger clearances are occasionally necessary to allow ball advance and retard because of misalignment or combined load (ref. 6). Standard land clearances can be used if the circumferential cage pocket clearance is sufficient to allow ball excursions.

Pocket clearance.—Cage-ball pocket clearance is extremely important in cases of misalignment or combined load (refs. 5 and 6). Pocket clearances of 0.025 to 0.035 in. are often used when some ball speed variation is expected. Elliptical cage pockets with major ellipse diameter in the circumferential direction are sometimes used when low transverse pocket clearance is desired to prevent cage wobble.

Cage sides protrusion.—Total cage widths are normally limited so that the cage side does not protrude beyond the bearing faces during operation. Countersunk or counterbored rivet-head sockets are often used instead of flat surfaces for the rivet heads to reduce cage width and still maintain the cage sidewall thickness that will provide adequate strength.

Cage rivet tightness.—Cage rivet fit and tension are carefully controlled to prevent loosening of rivets. Plating is not allowed between cage halves or under rivet heads. Rivet and cage materials are thermally matched to prevent loosening by differential response to changes in temperature. The diameters and lengths of cage rivets are established to obtain rivet stretch sufficient to maintain tightness while not deforming the cage or side plate material.

3. DESIGN CRITERIA and Recommended Practices

3.1 Bearing Assembly Design

3.1.1 Load Capability

The bearing shall be suitable for supporting the type and magnitude of the predicted loading (radial only, one-direction axial only, two-direction axial only, and combined radial and axial loads).

Bearings fulfilling the load direction requirements shown in table I should be analyzed for dynamic characteristics (including speed effects) by methods given in reference 3. The most favorable configuration and internal geometry should then be established on the basis of fatigue life, heat generation, stiffness, coolant suitability, and misalignment tolerance. It is recommended that bearings for turbopump service display a calculated minimum B_{10} fatigue life of 10 times the required turbopump life.

3.1.2 Speed Capability

The bearing shall satisfy the speed requirements of the application.

The use of any bearing type that dynamic analysis shows has either of the following weaknesses should be avoided: (1) an outer race inadequate to survive fatigue induced by centrifugal force of the rolling elements; or (2) a cage whose structural capacity is exceeded by centrifugal force or is subject to unsatisfactory dynamic interaction such as excessive sliding and heat generation of the rolling elements and races. The limitation on the last condition is difficult to define quantitatively because it is subject to improvement by changes in loading, cooling, and geometry.

Experience to date in the design of bearings for high-speed applications provides a basis for the following recommendations:

(1) One-piece cages should be used for speeds over 1.0×10^6 DN.
(2) Angular-contact ball bearings should be used for speeds between 1.0 and 3.0×10^6 DN.
(3) Reference 8 should be consulted for maximum speeds with cylindrical roller bearings.
(4) Rolling element bearings should not be used for speeds over 3.0×10^6 DN without prior testing.

3.1.3 Stiffness

Through a dynamic analysis of performance under operating conditions of speed and load, the proposed bearing design shall exhibit stiffness adequate for the intended use.

Digital computer programs based on the generalized bearing analysis presented in reference 3 should be used to analyze bearing designs; such programs provide a means for quickly performing complex calculations that would be prohibitively time consuming to perform by hand.

If the stiffness of a proposed design is inadequate, another design or loading condition must be evaluated. For example, assume that a proposed design of a single ball bearing has too low a radial stiffness; any of the following design alternatives will yield a stiffer bearing:

(1) Increase the size of the bearing.
(2) Increase the axial preload.
(3) Decrease the contact angle.
(4) Change the design to incorporate a greater number of smaller diameter balls.
(5) Substitute a duplex pair of ball bearings for the single bearing.
(6) Substitute a roller bearing for the ball bearing.
(7) Lengthen rollers (see sec. 3.2.1.5 for precautions on maximum roller length/diameter ratio).

3.1.4 Misalignment Tolerance

The misalignment capabilities of the bearing design shall exceed the misalignment imposed by the application.

The misalignment imposed on the bearing in the turbopump design should be minimized. The maximum recommended stationary race misalignment of common bearing types is presented in table V.

The misalignment tolerance in ball bearings can be maximized by providing adequate circumferential cage pocket clearance to allow for orbital ball speed variation (BSV). In reference 6 Barish discusses the cause and effects of BSV and proposes a means of calculating the cage clearance required to minimize cage distress caused by BSV.

A roller crown should be provided in roller bearings to reduce the loss of capacity caused by the edge loading of the rollers (see sec. 3.2.1.7).

TABLE V.—Bearing Misalignment Capabilities

Bearing type	Misalignment capability		
	(1)	(2)	(3)
Conrad-type (ref. 6)	0°15′	0°25′	—
Angular-contact, spring preload (ref. 6)	0° 2′	0°10′	—
Angular-contact, rigid preload	—	—	0°0′
Cylindrical roller bearing (ref. 20)	—	—	0°5′
Split-inner-ring (ref. 20)	0° 2′	—	—

Notes:

(1) Cage-guiding land and ball-ball pocket clearances: 0.005 to 0.015 in.

(2) Cage-guiding land and ball-ball pocket clearances: 0.025 to 0.035 in.

(3) No clearances specified.

3.1.5 Bore

The bearing bore shall be sufficiently small to reduce the detrimental effects of high speed.

The bearing with the smallest diameter that meets shaft strength and stiffness requirements should be utilized.

When analyzing the shaft stiffness to determine the minimum bearing bore, the stiffening effect of spacers, races, and clamping forces should be included; this effect may allow a smaller bearing size to be used.

The maximum allowable bearing bore is limited by the centrifugal loading on the materials of the rolling elements at design speed. A tandem set of ball thrust bearings should be used when the capacity of a single bearing is inadequate but an increase in bearing bore would result in a loss of capacity because of centrifugal loading.

Do not put a spline inside a bearing. A spline generally enlarges the required bore and may result in inner race taper distortion produced by spline tooth loads.

3.1.6 Internal Clearance

Internal clearance in rolling bearings shall not become negative.

The unfitted diametral clearance should be specified to maintain positive operating diametral clearance by using the formula

$$C_d = C_o + C_f + C_t + C_c$$

where

C_d = Unfitted diametral clearance
C_o = Operating diametral clearance, which should be (1) the results of the contact angle and curvatures, or (2) 0.0005 in. minimum
C_f = Clearance change caused by press fitting (see sec. 3.1.8)
C_t = Clearance change caused by the temperature difference between the inner and outer race. Measured values or heat-transfer analysis results should be used. If temperatures are unknown, use the following guideline values when the inner race is warmer than the outer race:

Bearing DN, millions	ΔT,°F
0.5	0
0.5 to 1.0	50
1.0 to 1.5	100
1.5 to 3.0	150

C_c = Clearance reduction caused by the centrifugal growth of the inner race (calculated by the methods of ref. 29, pp. 214-221).

Exceptions to the rule for maintaining positive diametral clearances should be considered in applications where roller skidding may occur because of insufficient radial load. Testing then must confirm that the cooling provided is adequate to establish stable safe temperature levels (ref. 8).

3.1.7 Cooling

3.1.7.1 Coolant Effects

The bearing type shall be suitable for and appropriate to the coolant available.

Ball bearings are preferable to roller bearings for nonlubricating coolant applications where the ball bearings are adequate for the required load capacity, speed capacity, and stiffness. Full-scale bearing tests should confirm the use of cylindrical roller bearings particularly in applications involving chemically reduced coolants.

The most favorable bearing design for propellant lubrication is the angular-contact ball bearing; the relieved shoulder provides good coolant circulation, and the separable feature of the bearing allows full cleaning and cleanliness verification that would be impossible with a Conrad-type bearing. Alternatively, a split-inner-ring bearing should be used where thrust load capacity is required in both directions.

3.1.7.2 Coolant Quantity

The flowrate of bearing coolant shall be sufficient to maintain temperatures that do not endanger bearing integrity.

Positive through-flow of bearing coolant should be of sufficient quantity to maintain acceptably low temperatures throughout the bearing system. The required quantity can be determined as follows:

(1) Estimate the heat-generation rate of the bearing by utilizing the results of dynamic analyses, or use an effective friction coefficient applied to bearing dimensions and load. For jet- or mist-lubricated applications involving oil and RP-1, use the following effective friction coefficients (refs. 30 and 31) applied at the bearing bore radius:

Bearing type	Rolling friction coefficient
Ball bearing, radial load	0.0015
Ball bearing, axial load	0.0018
Cylindrical roller bearing	0.0010

As bearing speed increases, the value used for friction coefficient should be increased to allow for churning; the increased value may be as much as twice the original value for jet-lubrication applications and up to ten times the original for flooded-lubrication systems.

For ball bearings with combined load, the conservative approach is to add the effects of the radial and axial loading. For close determination of the effective friction coefficient, test the actual bearing.

(2) Estimate the heat input from external sources such as turbine heat, heat from the atmosphere (for cryogenic coolants), or frictional heat from seals, gears, or other mechanical devices associated with the bearings.

(3) Determine the allowable heat capacity of the coolant.

For storable propellants, subtract the maximum expected inlet temperature from the maximum desirable fluid temperature (that which assures no degradation of the fluid) to obtain the minimum anticipated useable temperature differential.

Multiply the ΔT found by the specific heat of the fluid to obtain the heat capacity per unit of mass allowable.

For cryogenic coolants, determine the enthalpy change for constant pressure heating that will maintain desirable fluid temperature (preferably liquid).

(4) Divide the total heat input from (1) and (2) by the allowable heat capacity of the fluid from (3). The result is the minimum quantity (flowrate of coolant required).

3.1.7.3 Coolant Delivery System

The coolant-system configuration shall not result in excessive heat generation.

The choice between a jet-fed and a flooded-lubricant system should be based on avoiding excessive heat generation caused by churning of the fluid.

Flooded-lubricant/coolant systems should be chosen for propellant-cooled bearings. Most flooded-lubrication systems return the coolant to the pump low pressure area, so the flow of coolant does not represent a complete loss.

All passage areas should be maximized to avoid restriction caused by contamination.

The flow on the downstream side of bearing should be throttled. This is particularly desirable in cryogenic cooling systems to maximize pressure and thereby inhibit vaporization.

Cumulative foaming of a lubricant in a recirculating lube system can be prevented by selecting a foam-resistant coolant and by providing adequate stay time in the oil sump to allow gas escape. Testing may be required to obtain optimum balance between foam control and system compactness.

In a recirculating lubrication system, a scavenge pump should be provided with a capacity four times the lubricant flowrate.

The flow system should be designed with the fewest possible turns, crevices, and dead-ended areas to facilitate maintenance of liquid conditions throughout the bearing area.

The return point for the coolant should be chosen so that it causes the least disturbance to the pump hydrodynamic performance.

The coolant should be taken from the cleanest area, and when possible centrifugal force should be used to exclude particles from the flow system.

Jet or mist feed should be used for coolants other than the pumped propellants.

In conventional lubricant or RP-1 systems, ample coolant exit passages from the bearing area should be provided. A simple rule of thumb is to use four times the area required for liquid flow to allow for foaming.

In noncryogenic coolant systems, submergence of the bearing in stagnant coolant or preservative can be prevented by placing the drain at the lowest point of the system.

Positive through-flow of coolant can be ensured by employing a slinger adjacent to the bearing.

The jet should be placed as close to the bearing as practical to ease targeting. Large-diameter small-cross-section O-rings should not be used as static seals for lubrication jet rings; they become clipped easily during installation. Standard-cross-section O-rings should be used when possible.

3.1.7.3.1 Jet Targeting

Lubrication jets shall direct the coolant to the area between the inner race and the cage.

To ensure that the stream of coolant always enters the area between the inner race and cage, a tolerance stackup of all related parts should be performed. Correct targeting of the jet can be obtained by (1) shimming a bayonet type jet to strike a scribed line on plastic target or (2) controlling closely the jet angle and diameter tolerances for fixed lubrication jet rings (fig. 12). All possible reasons for mistargeting the stream must be eliminated. Jets should be checked for flowrate and for stream separations that would make a significant portion of the flow ineffective.

Outer-land-riding cages should be utilized to increase the target area (the gap between inner race and cage).

If the use of inner-land-riding cages is unavoidable, the bearing area should be pressurized to 2 psig minimum with gas (such as nitrogen) so that sufficient windage occurs to ensure distribution of the lubricant.

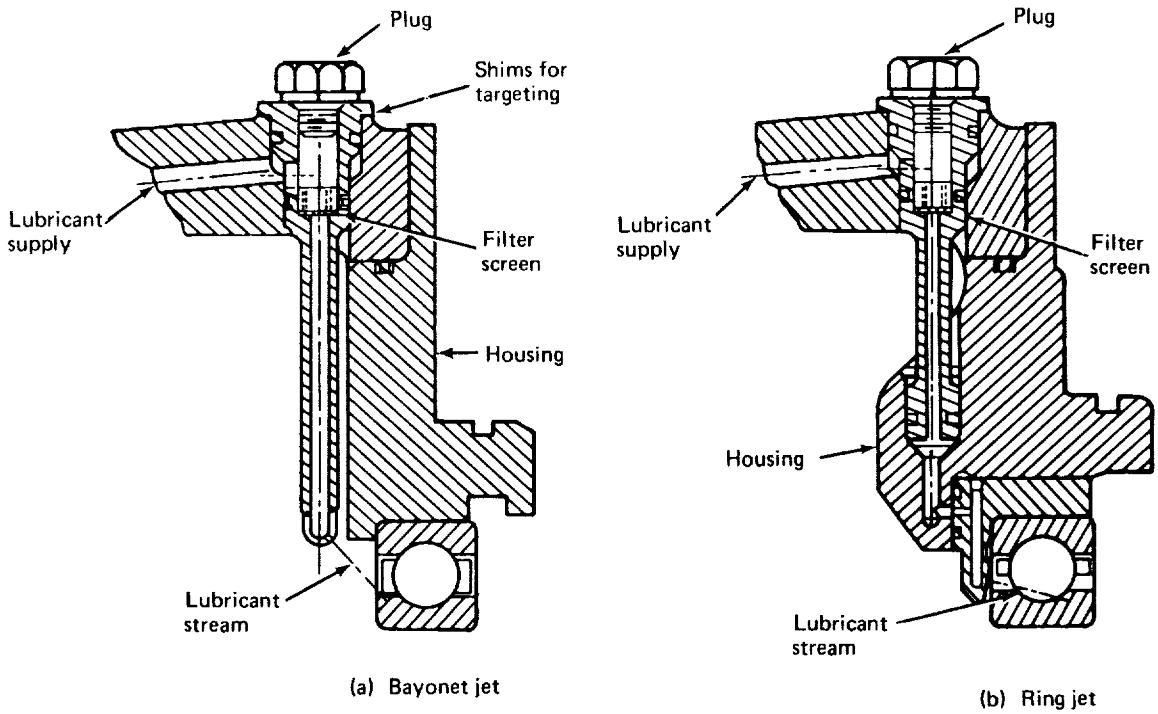


Figure 12.—Types of lubrication jets.

3.1.7.4 Coolant Quality (Liquid/Vapor Fraction)

For maximum cooling effectiveness, the coolant shall exist in the liquid state throughout the bearing area.

The cryogenic coolant pressure and temperature should be maintained to ensure a liquid state at the inlet to the bearing.

If the temperature of the fluid is above the critical temperature, the pressure of the coolant should be maintained above the critical pressure level (ref. 11).

3.1.7.5 Contamination

3.1.7.5.1 Chemicals

There shall be no material chemically incompatible with the coolant in any propellant-cooled bearing system.

Bearing components should be cleansed in nonpetroleum solvents before assembly. Chlorinated or fluorinated hydrocarbons may be used; with either, ensure adequate ventilation and minimum exposure of personnel.

The particle count and NVR (nonvolatile residue) requirements developed for liquid oxygen systems in specification MSFC-SPEC-164 (ref. 12), which controls the contamination per square foot of component, should be enforced as follows:

Particle size	Number allowed per ft^2
175 to 700 microns	5
700 to 2500 microns	1
$>$ 2500 microns	0
Nonvolatile residue	1 mg/ft^2 maximum

Passivation of all systems using fluorine or FLOX should be provided (ref. 32, pp. 212-317).

3.1.7.5.2 Water

No water shall be in the bearing area during idle periods.

Bearings cooled by RP-1 and conventional lubricants should be protected during idle periods by coating with water-displacing preservatives. Suitable preservatives should be similar to compounds that meet MIL-L-16173 (ref. 33). The preservative selected must be compatible with all materials it may contact (e.g., leaked propellant, products of combustion, etc.).

Water vapor should be excluded from propellant-cooled bearings by sealing the system from the atmosphere.

To prevent the entry of water and to remove water from propellant-cooled bearing cavities, purging with dry nitrogen is recommended. Dry helium gas should be used for purging liquid hydrogen systems. In cryogenic coolant systems, the purge must be maintained during chilled periods to prevent ice formation.

3.1.7.5.3 Particulate Matter

No particulate matter that might damage the bearing or plug the lubrication passages or jets shall contaminate the lubrication system.

The solvents used in cleaning bearings should be passed through 10μ filters. Preservatives and greases should be passed through filters with a minimum size opening of 40μ; finer filters may remove the active ingredients of preservatives. Coolant delivery passages should be equipped with screens or filters having a mesh size of 40μ or less. When designing lubrication jets, a provision for chip cleanout after machining should be included.

Press fits should not be used as static seals for lubrication jets or lubrication jet rings because metal chips shaved off during the pressing operation may later clog the jet orifices.

For systems subject to oil sludging from combustion products, replaceable filters are recommended. A bypass valve should be provided to avoid complete oil starvation should the filter clog.

3.1.8 Bearing Mounting

3.1.8.1 Interference Fits

The fit between the rotating race and its shaft (inner race rotation) or its housing (outer race rotation) shall be tight enough at all operating temperatures to prevent relative motion and consequent fretting, but shall not be so tight at any anticipated temperature that brittle fracture of the raceways results.

Care should be taken to obtain the minimum possible interference fit of the bearing race that will prevent relative rotation at operating temperature. The selection of room temperature shaft and housing dimensions should follow the steps outlined below:

(1) Determine the fit required at operating temperature. Choose the fit shown for the race rotation (inner or outer) and the bore size shown in figure 13 for inner races and figure 14 for outer races.
(2) Calculate the effect of differential contraction or expansion on the operating temperature of the shaft and housing materials.
(3) Determine the interference necessary to prevent creeping caused by the Poisson effect of rolling element loads (ref. 31, pp. 122-123).
(4) Use integral races where possible if extra-heavy interference fits are required.
(5) Calculate the inner ring tensile stress resulting from the interference fit for both ambient and operating temperatures. Use the methods of reference 34, Vol. II, pp. 161-170. Modify the fit to reduce the interference if, under any temperature condition, the tensile stress approaches the strength of the race material. Consider that race materials have very small elongation before fracture.

3.1.8.2 Sliding Fits

Spring preloaded races requiring axial freedom to maintain preload shall have sliding fits at all operating temperatures.

Outer races of spring preloaded bearings should be fit at operating temperature as shown in the lower curve of figure 14. Differential contraction or expansion should be compensated for to determine manufacturing dimensions.

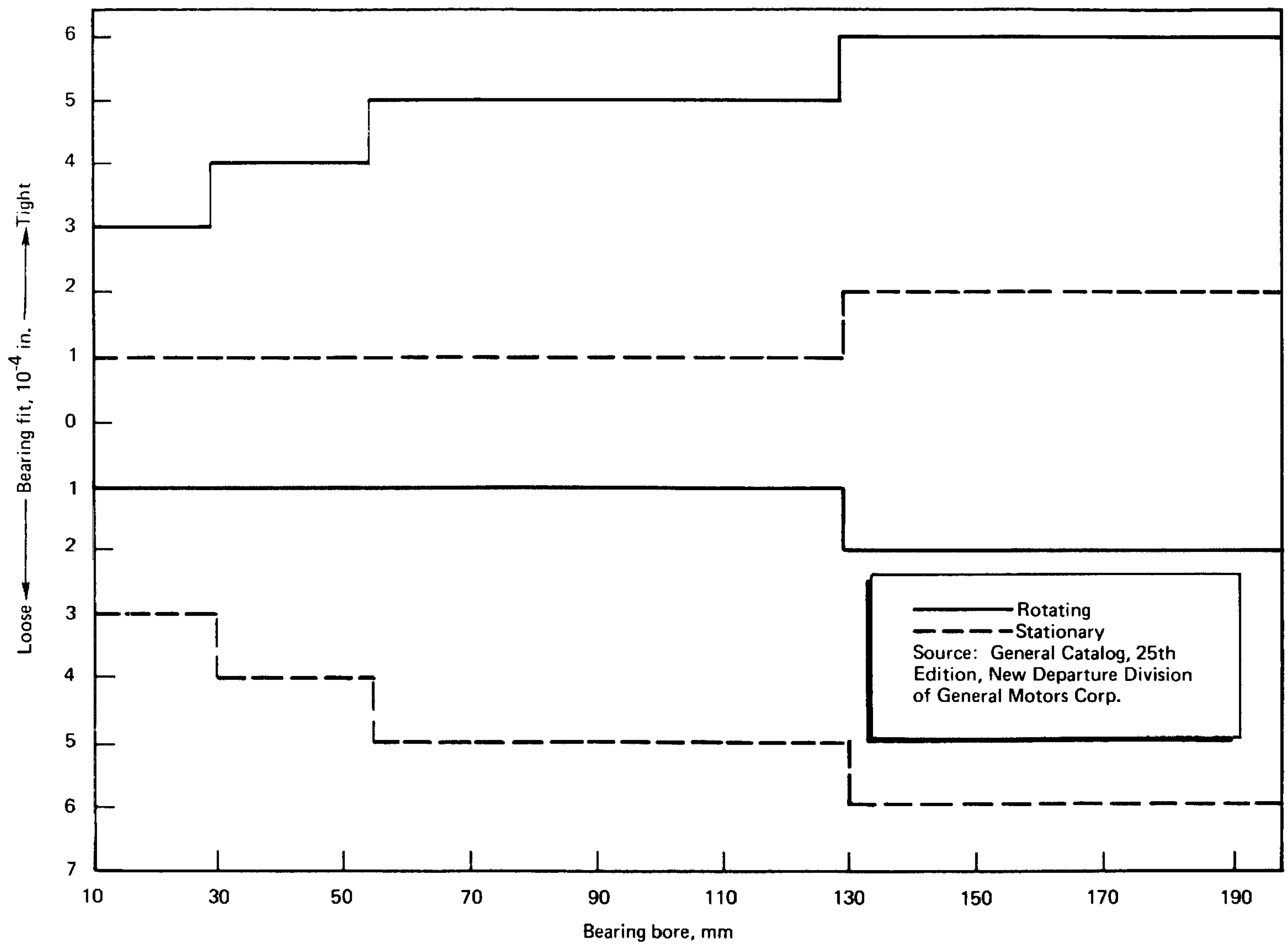


Figure 13.—Inner race fits, ABEC-5.

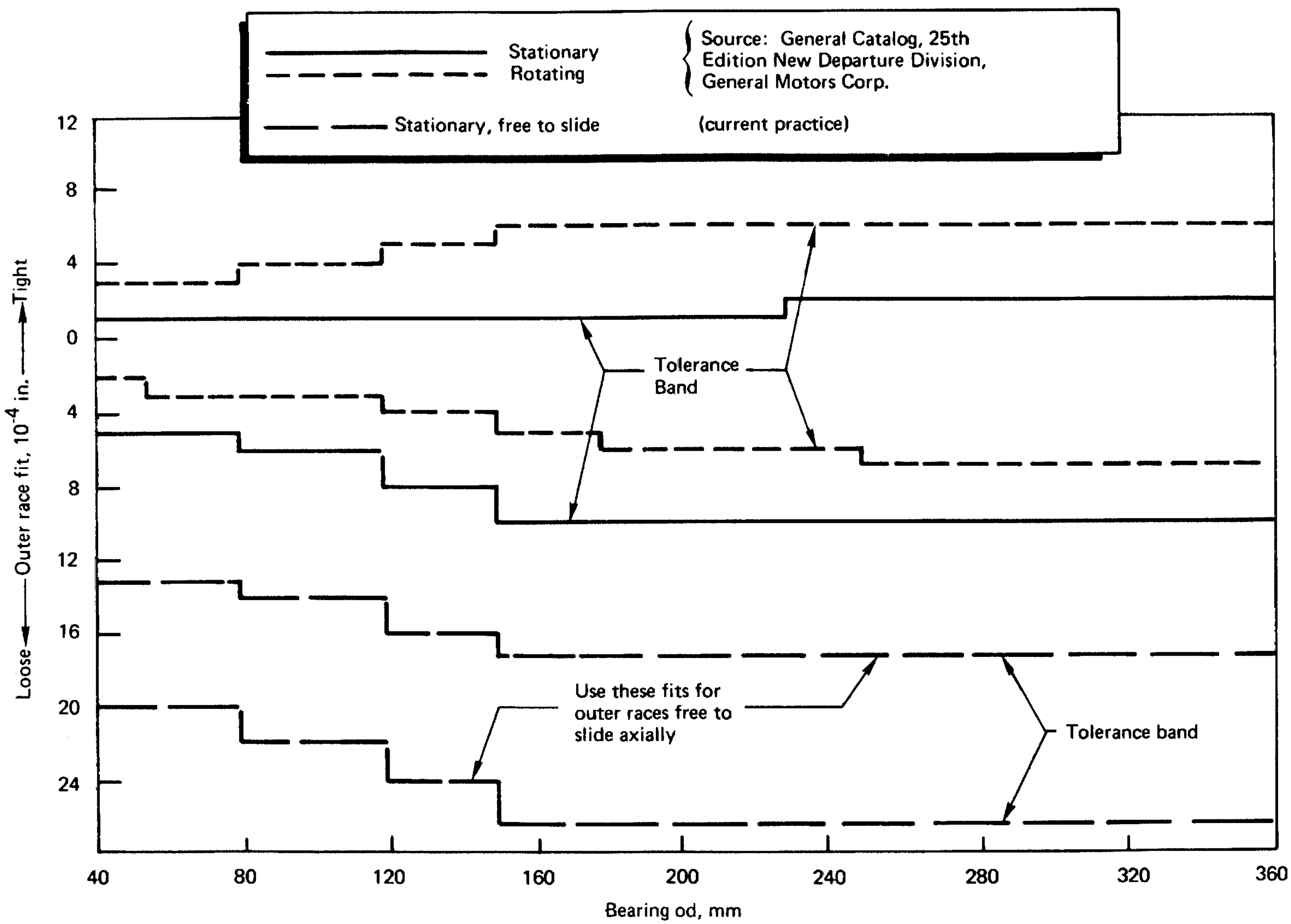


Figure 14.—Outer race fits, ABEC-5.

3.1.8.3 Bearing Mounting Surfaces

Shaft journals and housing bores shall not be subject to galling and fretting.

Galling and fretting of shaft journals and housing bores that may result from repeated assembly or from relative motion of axially free races should be prevented by one of the following practices:

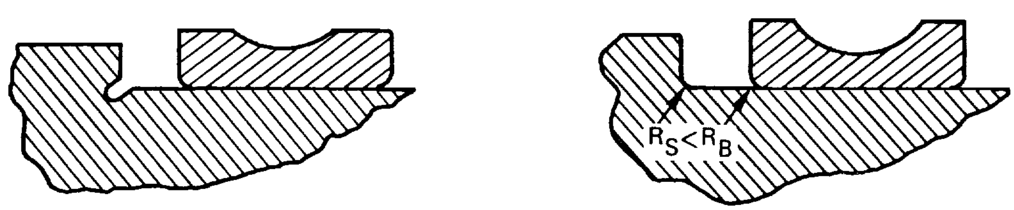
(1) Chrome plate journals and bores. The recommended practice is to plate with hard dense (not decorative or porous) chrome per Federal Specification QQ-C-320 (ref. 35). The plating should be thick enough to attain 0.002 to 0.006 in. of chrome after grinding. Finish grind the plating to a finish of 16μ in. AA. This practice is essential for journals and bores made of Inconel and Monel to prevent galling with 440-C bearing rings.
(2) When using 4130, 4340, or 9310 steel for shafts and housings mounting bearings made of SAE 52100 and SAE 4620, harden journals and bores to 40 R_c minimum.
(3) When titanium shafts are used, install the bearing with a light press fit.

3.1.8.4 Corner Radii Interference

Shaft shoulder corner radii shall not interfere with raceway corner radii.

To ensure that the shaft shoulder radii do not interfere with the raceway corner radii one of the following practices should be utilized:

(1) Use a grind relief (fig. 15(a)) instead of corner radius at the shaft shoulder (fig. 15(b)) when this does not cause an excessive concentration of shaft stress.



(a) Shaft shoulder with grind relief

(b) Shaft shoulder with corner radius

Figure 15.—Shaft shoulder design.

(2) Specify that the corner radius of shaft R_S shall be smaller than the minimum bearing corner radius R_B (fig. 15(b)).

3.1.8.5 Mounting Surface Accuracy

The concentricity and normality of the bearing mounting surfaces shall be consistent with and adequate for the bearing alignment requirements.

Shaft diametral tolerances should be compatible with the class of precision of the bearing specified (ABEC 5, RBEC 5, or ABEC 7).

A maximum taper control of 50μ in. per inch should be specified for interference-fitted journals and housing bores.

A surface finish of 16μ in. AA should be specified for bearing journals and housing bores.

The concentricity of housing bores and normality of housing shoulders that will result in a maximum misalignment, including runouts of faces, should be specified as follows:

(1) Conrad-type bearings with loose cage or no thrust—0° 25′
(2) Conrad-type bearings or angular-contact bearings with loose cage—0° 15′
(3) Thrust-load-carrying ball bearings and cylindrical roller bearings—0° 5′

Shaft shoulder normality should be compatible with the bearing tolerance. The following values should be used as guidelines:

Bearing bore, mm	Shoulder normality, inches TIR
0 to 80	0.0003
80 to 120	0.0004
120 to 200	0.0005

3.1.8.6 Shoulder Heights

Shaft and housing shoulder heights shall be sized for sufficient axial support of the race, and shall allow removal of the bearing.

When using commercially available bearings, shaft and housing shoulder heights should be chosen from the heights recommended in the manufacturers' catalog. For most rocket engine turbomachinery applications, the guidelines below should be used:

(1) The shaft or housing shoulder should be larger than the maximum bearing race corner radius (fig. 16). Conversely, the corner bearing race corner radius should be smaller than the minimum shoulder height.
(2) The contact area between the shaft shoulder and race face should be sufficient to keep the compressive stress below the compressive yield strength of the shoulder material under clamping thrust loads.

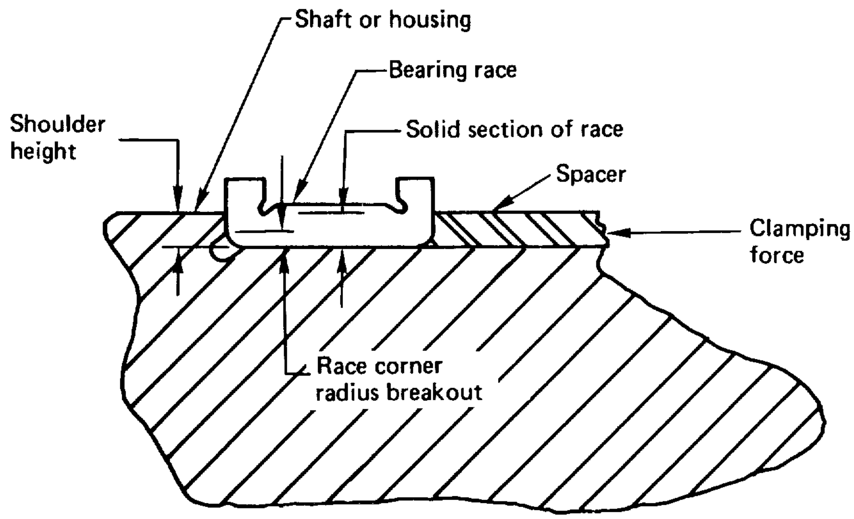


Figure 16.—Housing or shaft shoulder height sizing.

(3) Shoulder diameters that will put the clamping load through the solid race section should be chosen; any condition that may result in shoulder bending should be avoided (see fig. 16).

(4) The shoulder should be smaller than the race face by a sufficient margin for puller tool contacts (fig. 17(a)) or another means of bearing removal, e.g., puller grooves in races (fig. 17(b)) or jack screw holes in flanges, should be provided.

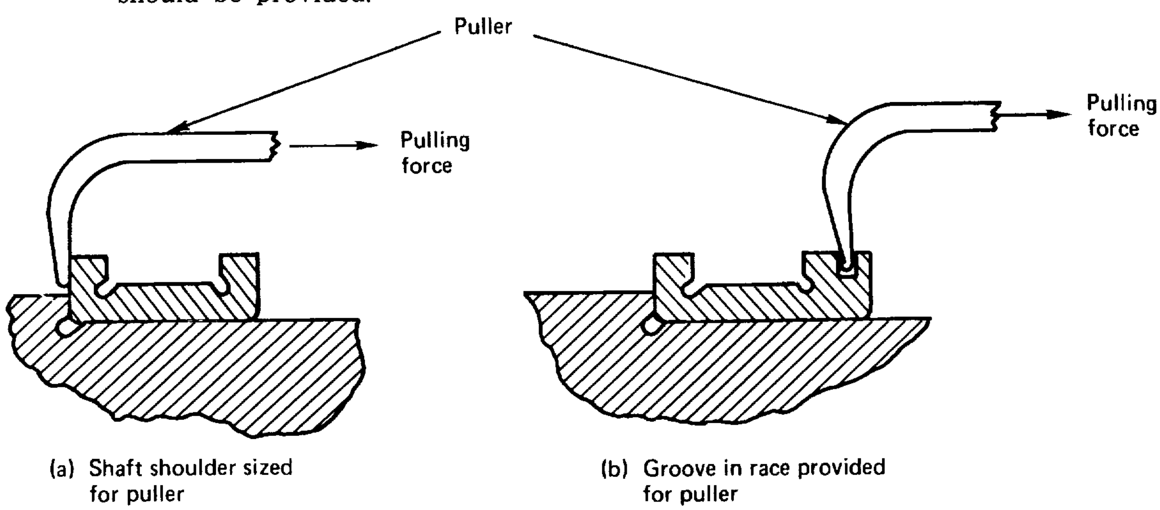


Figure 17.—Provisions for race removal.

3.1.8.7 Race Retention

Race retention methods shall maintain proper support and clamping loads for all thermal and loading conditions including friction, vibration, and shock.

The elastic deflection imposed by clamping devices should be sufficient to maintain a clamping force for any loading that will occur over the thermal range expected in the application. Snap rings should be avoided. The required nut torque should be calculated from the thread dimensions and force relations shown in reference 36, section 3, p. 48.

Bearing retaining clamping loads should be set so that the compressive stress exceeds the tensile bending stress induced in the shaft caused by bending (fig. 18). Failure to accomplish this will result in heavy fretting of the mating surfaces (ref. 7).

3.1.8.7.1 Clamping Loads

Clamping loads applied to bearing races with heavy rotating loads shall be sufficient to prevent both race creeping and fretting resulting from the Poisson effect of the individual rolling element loads and loosening of the stackup of parts resulting from bending of the shaft.

The clamping load required to prevent creeping caused by the race-widening Poisson effect of concentrated compressive loads at the rolling element positions should be determined by experiment. This clamping load may be impossible to attain at extremely high radial loads; the only solution then is to utilize integral inner races, which completely eliminates the problem.

Locking devices, e.g., bendup washers or tabs bent into slots to prevent loosening of clamping nuts, should be included. Tabs should be bent in the same direction as the centrifugal force on the rotating locks.

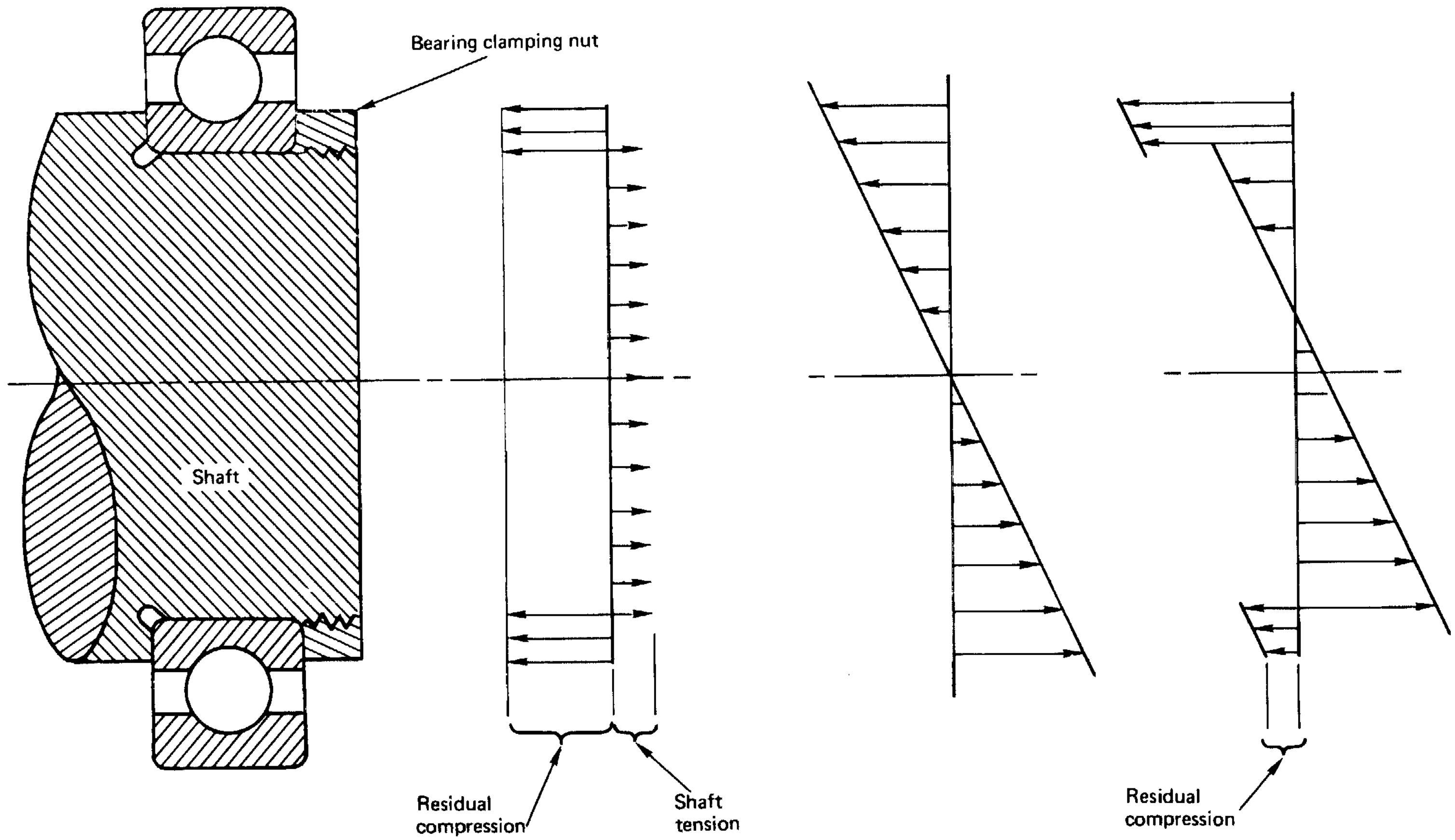
3.1.9 Bearing Materials

3.1.9.1 Corrosion Resistance

The rolling element, race, and cage materials shall be resistant to or protected from corrosion under all operating conditions.

AISI 440-C corrosion resistant steel is recommended for most propellant-cooled bearing applications. While no known material ideally fulfills the requirements of hardness, ductility or toughness, machinability, and availability, 440-C best meets the requirements with the fewest limitations.

Other bearing materials should be considered for propellant-lubricated applications where more complete corrosion resistance is required. Table VI presents a few of the possible materials choices, with some properties and disadvantages.



(a) Bearing inner race clamped through sleeve

(b) Shaft and sleeve forces

(c) Bending moment

(d) Resultant showing residual sleeve compression

Figure 18.—Required clamping force.

TABLE VI.—Corrosion Resistant Race and Rolling Element Materials

Temperature range, °F	Designation	Hardening	Maximum hardness, R_c	Disadvantages
−423 to 700	440-C	Through	60	Brittleness; not completely corrosion resistant
−423 to 1200	Haynes 25	Work hardened	55	Low hardness; limited availability; high cost
−425 to 68*	Stellite Star-J	Cast chill cast	52 61	Lack of test experience; high cost
	Stellite 19	Cast chill cast	49 55	Low hardness
	Stellite 3	Cast chill cast	51 59	Low hardness
−423 to 68*	Titanium Carbide K162B	Through	76	Brittleness; high modulus resulting in high stresses; high cost; difficult to fabricate
−423 to 68*	Tungsten Carbide	Through	76	Brittleness; high modulus; heavier than steel

*For hot hardness, the manufacturer's data should be consulted, then capacity adjustment should be applied according to the relationship given in section 3.1.9.2.

For RP-1 or conventional-lubricant cooled bearings, the use of standard bearing materials is recommended, with a provision for corrosion protection with preservatives. Materials that should be considered are

(1) SAE 52100
(2) Tool steels: M-10 and M-50
(3) SAE 4620 carburizing
(4) Proprietary steels as recommended by bearing suppliers

3.1.9.2 Hardness

Rolling element and race materials shall possess sufficient hardness to prevent plastic deformation under all loading conditions.

The capacity rating of bearing steels based on their hardness at operating temperature converted to tensile strength should be determined according to the relationship

$$\text{Capacity} \approx (\text{ultimate tensile strength})^n$$

where

$n = 3$ for ball bearings
$n = 2$ for roller bearings
$2 < n < 3$ for crowned roller bearings

See reference 37 for a more detailed discussion of hardness effect on capacity of rolling surfaces.

Fatigue properties of nonferrous bearing materials (e.g., sintered carbides, ceramics, nickel alloys) should be determined by test.

To obtain maximum fatigue life, a differential hardness that makes rolling elements harder than the races by 1 to 2 points R_c (ref. 38) should be specified. Inherently, balls will be a few points harder than races because of the effect of smaller mass in heat treatment.

3.1.9.3 Adhesion Resistance

The rolling element and race materials shall resist adhesion during rolling contact in the presence of the selected coolant.

440-C CRES is suitable for use as a race and ball material with the following propellant/coolants (ref. 18):

Fuels	Oxidizers
RP-1	Liquid oxygen
Liquid hydrogen	IRFNA
B_5H_9	Liquid fluorine

Tests should be conducted under simulated operating conditions before using 440-C in bearings cooled by the following propellants:

N_2H_4 (hydrazine)
Ethylene diamine
UDMH
50 percent UDMH–50 percent N_2H_4

3.1.9.4 Dimensional Stability

Raceway materials for cryogenic bearings shall not change dimensionally during use.

To obtain dimensional stability for 440-C races, repeated chilling and tempering cycles should be performed before final finishing operations.

A recommended stabilization procedure for 440-C intended for cryogenic service is

(1) Temper to +325° to +350° F.
(2) Cold soak in liquid nitrogen (−320° F) for 30 minutes minimum.
(3) Temper again to +325° to +350° F before final grinding.

3.1.9.5 Grinding Burns

Raceways shall not be subject to grinding burns.

Grinding coolant flow, feed rate, and grinding wheel composition should be controlled by the manufacturer to preclude grinding burns in raceways. Suspected grinding burns can be detected with Nital etch on carburizing steel. Ensure that the acid is neutralized and that surface effects are removed by final finishing.

3.1.9.6 Cage Materials—Friction

Sliding friction of cage materials against rolling elements and races shall not result in excessive forces and heat generation.

Table VII presents cage materials recommended for use because they display good frictional characteristics with the indicated propellant, i.e., the coefficient of friction is between 0.05 and 0.15.

Bearing cages for bearings cooled by RP-1 or conventional lubricants should follow aircraft bearing practice, as follows:

Temperature range, °F	Material
−65 to 250	Linen-supported phenolic resin
−65 to 600	Silver-plated iron silicon bronze
	Silver-plated 4130 or 6414 steel
−65 to 1000	Oxide-coated S-Monel

TABLE VII.—Recommended Cage Materials

Oxidizer	Fuel	Cage material				
		Armalon[2]	Salox M[3]	Rulon A[4]	25 percent GFT[5]	K-Monel
	Amines[1]				X	
	B_5H_9	X				
	Liquid hydrogen	X	X	X	X	
	Gaseous hydrogen		X	X		
N_2O_4					X	
IRFNA		X				
Liquid oxygen		X	X	X		
FLOX[6] or liquid fluorine						X

[1]N_2H_4, UDMH, or 50-50 mixture
[2]Mandrel-wrapped, glass-fabric-supported PTFE
[3]40 percent bronze-powder-filled PTFE
[4]15 percent glass-fiber-filled PTFE
[5]25 percent glass-fiber-filled PTFE
[6]Mixture of liquid fluorine and liquid oxygen

3.1.9.7 Materials Cleanliness

Materials used for races and rolling elements shall be free of nonmetallic inclusions.

Consumable-electrode vacuum-melted materials should be specified for race and rolling elements.

3.1.10 Testing

Bearing tests shall confirm the adequacy of the design.

Full-scale testing.—Full-scale bearing tests simulating operating environment should confirm design adequacy prior to production use of the bearing. Test sequences should confirm bearing life and structural integrity over the entire range of turbopump speed and load conditions. The dimensions of the test bearing should cover the range of manufacturing tolerances, particularly diametral clearance.

Rolling contact tests.—Modified Shell Four-Ball (Barwell) Tester (ref. 19), NASA Five-Ball Tester (ref. 38), or similar subscale testers should be used to determine the rolling contact performance of candidate materials for untried coolants.

Sliding friction tests.—The use of simple sliding tests, e.g., a button on a rotating disk (ref. 23), is recommended to shorten significantly the search for low-friction, long-wearing cage materials.

The results of subscale rolling and sliding friction tests will indicate the most fruitful avenues of investigation for full-scale bearing tests, thus shortening the total design cycle.

3.2 Bearing Component Design

3.2.1 Rolling Element Design

3.2.1.1 Size

The ball or roller diameter shall be sufficiently large to obtain the required bearing capacity, adequate cage radial thickness, and adequate coolant flow area; the diameter shall be less than that which reduces capacity because of increased centrifugal force.

The number and size of rolling elements should be optimized by selecting from iterated dynamic analyses the most favorable of candidate bearing design geometries. Bearing manufacturers' catalogs represent optimized dimensions for most conditions and are therefore a good starting point for special design optimization. Figure 19 presents the approximate ball diameter/pitch diameter ratio versus bore for the various cross-section series for standard ball bearings; figure 20 presents the same information for roller bearings.

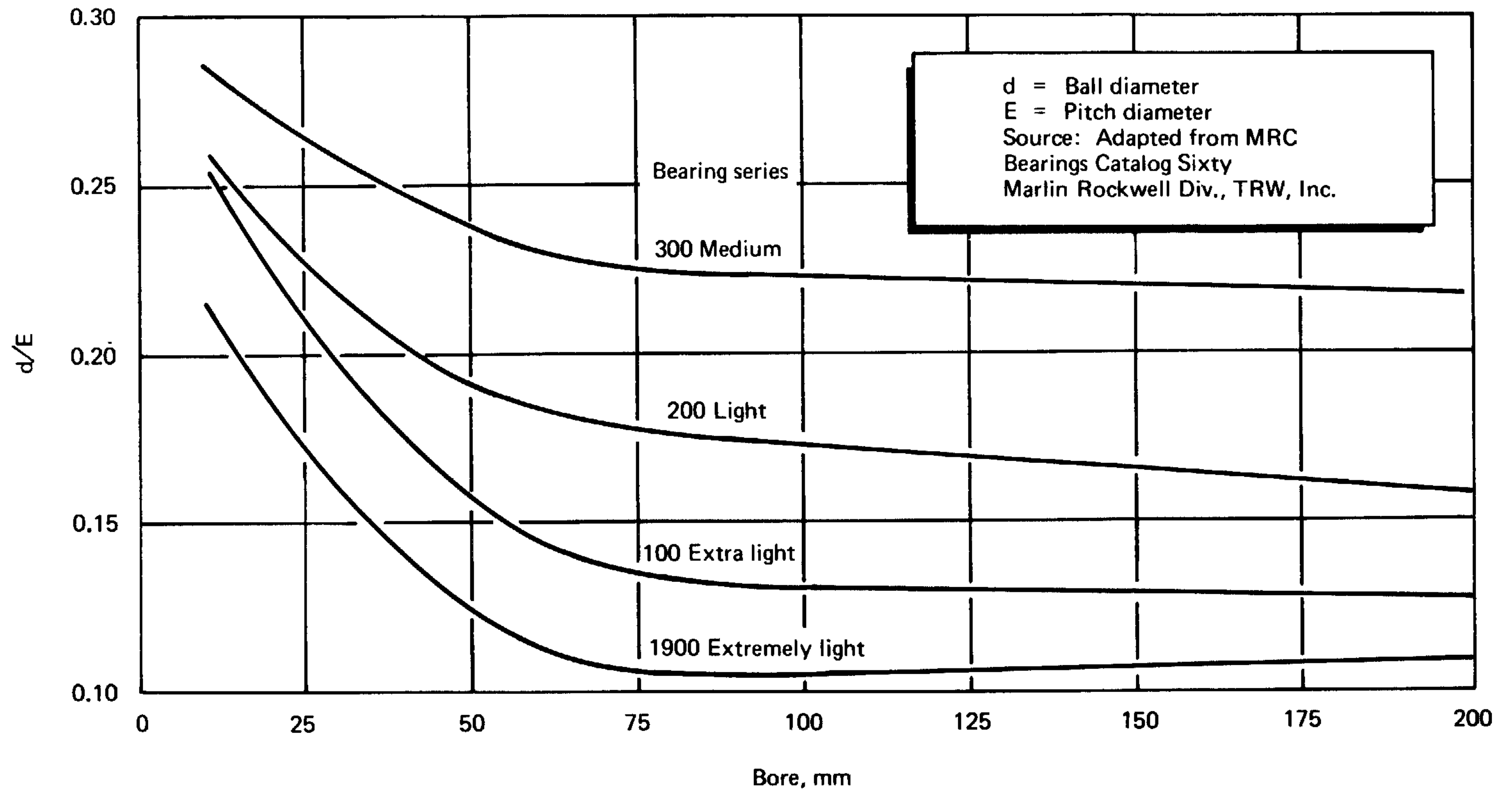


Figure 19.—Ball size basic proportions.

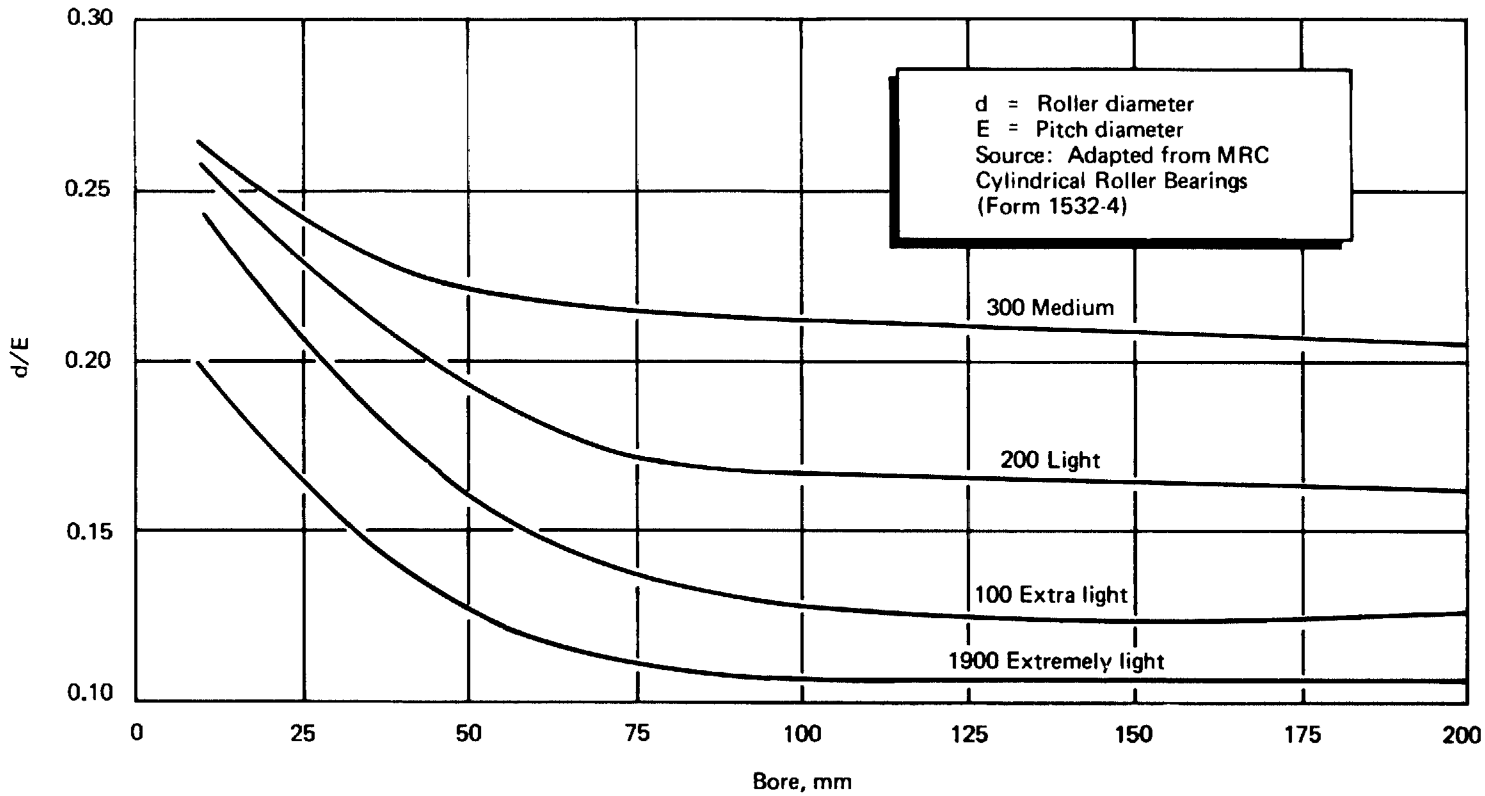


Figure 20.—Roller size basic proportions.

Maximum ball size.—Since centrifugal force varies as d^3 and capacity varies as $d^{1.8}$, the maximum desirable ball size will result. An acceptable rule of thumb proposed by Barish is to limit the ball size so that the following inequality holds:

$$\frac{\text{Sum of ball centrifugal forces}}{\text{500 hr capacity at speed}} \leq 2 \text{ for air-melt steels, or } \leq 2.88 \text{ for vacuum-melted steels}$$

If finer determination of maximum ball size is required, iterated calculations of fatigue life should be made for proposed bearing designs varying ball size and number (see sec. 3.2.1.2); the ball size that results in the required life of the bearing should be selected.

Minimum ball/roller size.—In minimizing rolling element size, sufficient size must be maintained to provide both structure for a cage of adequate strength and an area adequate for through-flow of coolant. In lieu of test experience to the contrary, the following guidelines are suggested for minimum proportions:

(1) Minimum cage radial thickness > 0.090 for nonmetallic cages

(2) $\frac{\text{Minimum cage radial thickness}}{\text{Cage average diameter}} \geq 0.015$

(3) $\frac{\text{Minimum open area for through-flow}}{\text{Bearing pitch diameter}} \geq 0.03$

Smaller dimensions should be subjected to development testing.

3.2.1.2 Number of Rolling Elements

The number of rolling elements shall be sufficient to give smooth bearing operation, but shall not exceed the quantity that results in the minimum allowable cage web thickness.

The maximum number of rolling elements should be determined so that the cage web thickness is not less than 0.150 inch for nonmetallic cages nor less than 0.070 inch for metallic cages. If the wear life of the cage is shown by testing to be insufficient, the number of rolling elements should be reduced.

The usual goal in bearing design is to maximize the number. If minimization is required for a special consideration, a rule of thumb is that the spacing between rolling elements should not be greater than the diameter of an individual element. To evaluate the effect of a minimum number of elements, the radial displacements caused by passing the elements through the radial load zone should be calculated, and their effects on shaft dynamics should be determined. For axial load, the polygonal distortion of the outer race (ref. 39, case 9, p. 158) should be calculated, and the harm that may result from the deflections should be determined.

3.2.1.3 Ball and Roller Diameter Uniformity

The ball or roller diameters within a bearing shall be sufficiently uniform to prevent ball speed variation and provide loading uniformity among individual balls or rollers.

Ball diameter tolerances should be specified as follows:

Bearing DN, millions	AFBMA grade balls	Maximum out of round, μ in.	Maximum size variation, μ in.
0 to 0.75	25	25	50
0.75 to 1.5	10	10	20
1.5 to 3.0	5	5	10

Where extremely smooth operation is required, the finer grades should be specified.

Specifications for roller size uniformity and cylindrical roundness should be equivalent to the AFBMA standards for balls.

3.2.1.4 Ball and Roller Surface Finish

The surface finish of rolling elements shall be fine enough to produce adequate load distribution in the contact area.

Ball finishes should be controlled by specifying the appropriate AFBMA grade. Roller outside-diameter maximum roughness should be specified as 4μ in. AA, and the roller ends as 6μ in. AA.

3.2.1.5 Roller Guidance

Roller guidance shall be adequate to prevent skewing and consequent roller end wear.

For roller bearings with diametral clearance, roller end play in the guiding lips must be maintained at a positive low value. The minimum end play should provide some allowance for thermal growth of the roller; the maximum end play should be limited to prevent a large roller skew angle. Suggested limits are 0.0005 to 0.002 in., with uniform length within 0.0002 in. for all rollers in one bearing assembly.

For specialized roller bearing applications, tests will be required to determine end play limits (ref. 8).

Shoulder height should be specified to contact the roller at approximately 75 percent of its diameter excluding corner radii. Suggested limits for shoulder height are 14 to 20 percent of the roller diameter.

Roller length/diameter ratio should be near unity for the best guidance. For a compromise between adequate stiffness and roller guidance, a maximum l/d ratio of 1.23:1 (ref. 40) should be used.

Parallelism of race guiding lips should be within 0.0002 inch.

Roller ends should be specified as square within 0.0001 in. TIR to prevent wear at the high points and aid in roller guidance.

3.2.1.6 Roller Corner Radii

3.2.1.6.1 Size

Roller corner radii shall be small enough to result in sufficient roller end contact bearing area but large enough to avoid interference with race corner radii.

Maximum roller end radii breakout height (b, fig. 21) should be less than one-half the shoulder height (h, fig. 22(a)) to preserve the roller end bearing area. Roller corner radii (r, fig. 21) should be greater than the race shoulder corner radius (R, fig. 22(b)).

3.2.1.6.2 Runout

Roller corner radii shall not cause unbalance of roller.

Roller corner radii runout should be controlled to 0.002 in. TIR to prevent excessive unbalance in high-speed bearings.

3.2.1.7 Roller End Relief (Crowning)

Roller end relief shall prevent compressive stress concentration at the ends of the rollers because of maximum anticipated misalignment.

Crown drop (e, fig. 21) should be greater than the elastic deflection of the heaviest loaded roller. The elastic deflection should be calculated using the method shown in reference 41. Crowning plus blending (as shown in fig. 21) should be used to prevent stress concentrations.

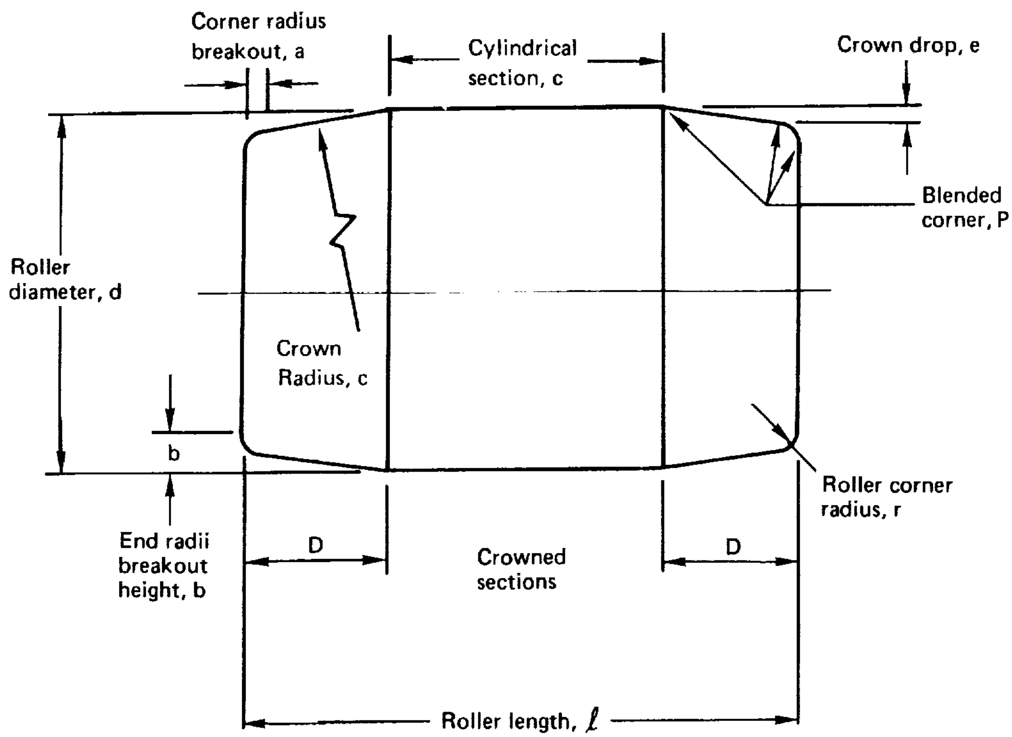


Figure 21.—Roller dimension terms and symbols.

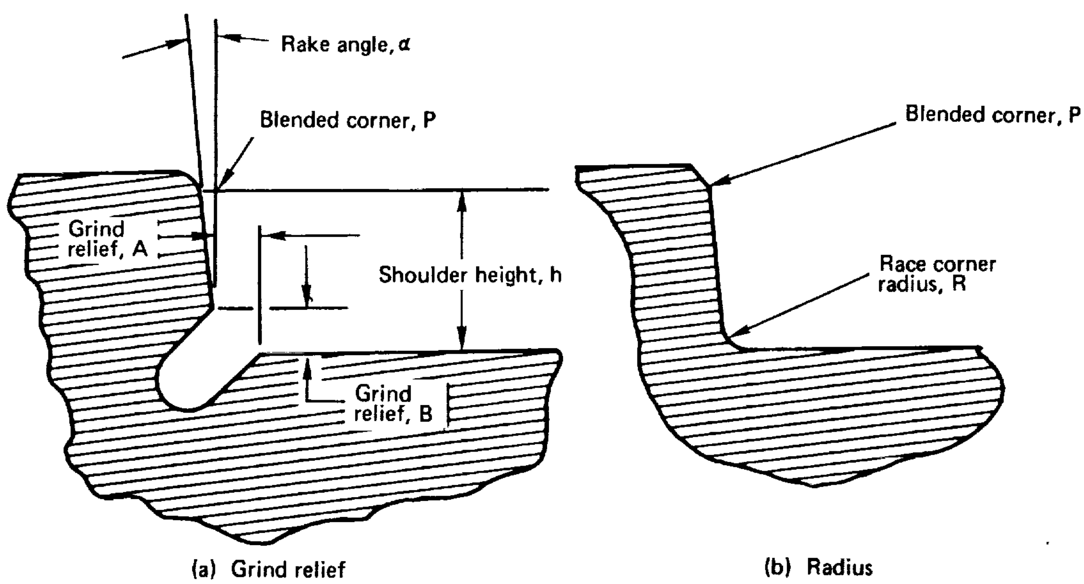


Figure 22.—Shoulder configuration terms and symbols.

If misalignment beyond 0°5′ exists in the application, the methods given by Harris (ref. 41) should be used to calculate the crown drop required to avoid edge loading. Blending of all such boundaries should be specified as shown in figure 21.

If no design control of roller crown is available, the dynamic capacity of the bearing should be derated as indicated in table VIII.

TABLE VIII.—Roller Bearing Capacity Reduction Caused by Bearing Misalignment*

Location misalignment, in./in.	Percent of basic dynamic capacity	Deflection misalignment, in./in.	Percent of basic dynamic capacity
0.0003	99.4	0.0003	97.5
0.0005	99.1	0.0005	96.0
0.0007	98.6	0.0007	94.4
0.0010	98.0	0.0010	91.8
0.0020	95.6	0.0014	88.3
0.0030	93.1	0.0018	84.2
0.0040	90.6	0.0022	79.6
0.0060	84.8	0.0026	74.4
0.0080	77.9	0.0030	67.9
0.0100	69.2		

*From Catalog AR-59, Rollway Bearing Co., Inc., Syracuse, N. Y., 1959.

3.2.2 Race Design

3.2.2.1 Cross Section

The race cross section shall be sufficiently massive to prevent brittle fracture of the race.

Races should be proportioned so the ratio of raceway thickness to mean raceway diameter does not fall below 0.05. For bearing raceways with lands (fig. 23) the average diameter D_{avg} and average cross-section radial thickness t_{avg} should be used.

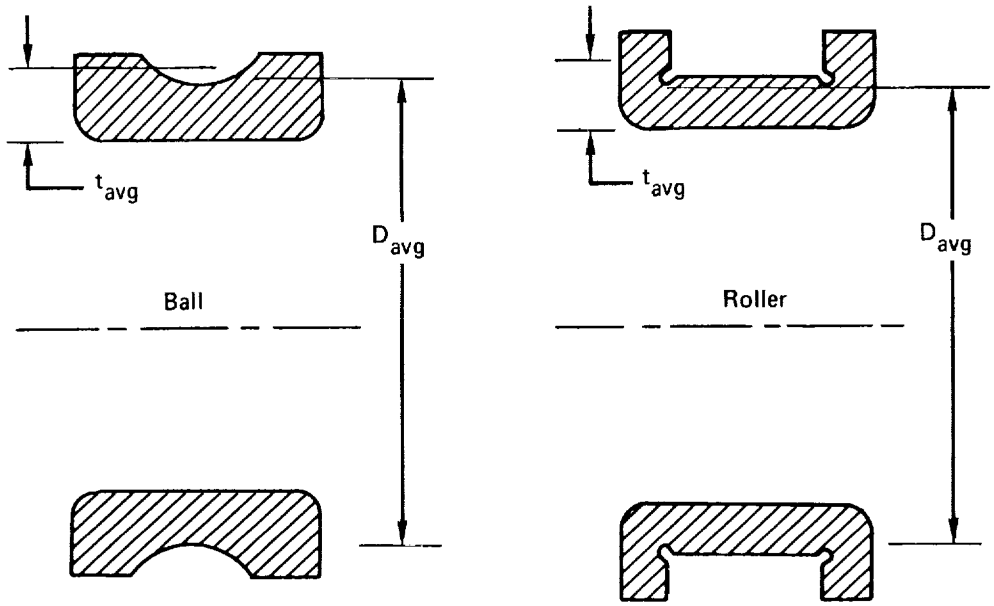


Figure 23.—Bearing race average diameters.

3.2.2.2 Ball Bearing Transverse Race Curvature

Race radii (curvatures) shall conform to ball radii closely enough to achieve the required load capacity, and yet not so closely that excessive nonrolling action (spinning and sliding of the balls) will occur and result in excessive heat generation and wear.

Race curvatures must be combined with ball size and the required diametral clearance to obtain the required fatigue life with acceptable ball spin. See reference 37 for the effect of sliding on fatigue life.

Determine whether the basic goals of the bearing design are minimum rolling friction torque or maximum capacity. For speeds in excess of 1.5 x 10^6 DN, bearings should be designed for minimum torque.

Perform dynamic analyses of candidate bearing designs, using a range of values for race curvatures. Suggested limits are 0.51 to 0.60. Select that bearing design with longest fatigue life or minimum friction torque consistent with the primary design goal.

For minimum friction torque, maximize curvatures with the limitation that adequate fatigue life should be maintained. Larger curvatures will result in smaller contact areas, less heat generation, but higher contact stress; consider that fatigue life varies inversely with the ninth power of stress.

If maximum capacity or high fatigue life is the objective, curvatures should be minimized but the minimum race curvature must not be so small that an excessively large contact angle will result from the requirements for diametral clearance (see sec. 3.1.6). A large initial contact angle will result in a large contact-angle divergence ($\beta_i - \beta_o$) under high-speed operation (ref. 3) resulting in excessive ball spin, heating, sear, and surface degradation. For static geometric relationships, see reference 34, Vol. I, pp. 1-4.

The race curvatures F_o (outer race) and F_i (inner race) should be varied independently to achieve the optimum overall effect for high-speed bearings (table IV).

For optimum high-speed bearing design, the minimum total curvature ($F_o + F_i - 1$) should be chosen so that initial contact angle is minimized, while the required diametral clearance (see sec. 3.1.6) is preserved. Minimum initial contact angle will result in minimized dynamic contact-angle divergence ($\beta_i - \beta_o$), and higher bearing radial stiffness.

IRC should be obtained for the specific bearing design, speed, and load by choosing a curvature combination F_o/F_i (refs. 25 and 26) so that $F_i < F_o$. It should be recognized that IRC is very difficult to achieve consistently with large bearings (50 mm bore) and high-speed bearings (DN $> 1.3 \times 10^6$). Whenever possible, however, the benefits of IRC should be obtained by using curvatures chosen as follows:

Curvature	Maximum value	Minimum value
F_o	That giving adequate fatigue life	That resulting in desirable β_o, P_D and equal fatigue lives for individual races
F_i	That required for IRC	

3.2.2.3 Race Finish

Raceway finish for ball or roller bearings shall be sufficiently smooth both circumferentially and axially to assure predictable stress distribution and smooth operation.

The currently acceptable race finishes provided in section 2.2.2 should be specified. Finer finishes should be considered if elastohydrodynamic (EHD) lubrication is to be used (ref. 28, pp. 36-39).

3.2.2.4 Ball Bearing Race Shoulder Configuration

The shoulder configuration for a ball bearing race shall prevent damage to balls under all normal loading conditions.

Using the results of the dynamic analysis, the minimum race shoulder height (fig. 24) should be specified large enough to contain the ball contact ellipse at all loading conditions and should provide a 50-percent overload safety factor. As an added precaution, an edge relief should be provided at the race shoulder as shown in figure 25.

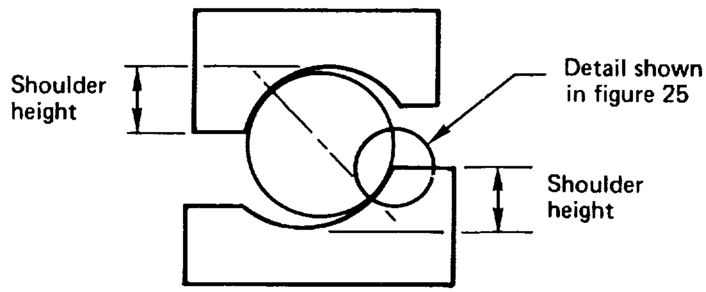


Figure 24.—Ball bearing race shoulder height.

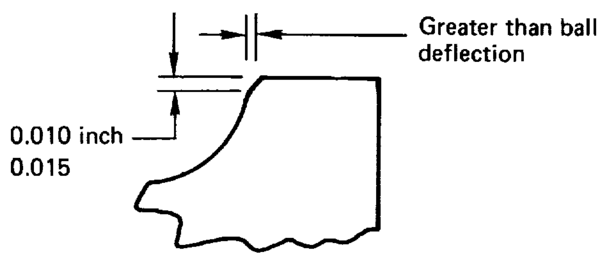


Figure 25.—Ball bearing raceway edge relief.

3.2.2.5 Roller Bearing Shoulder Configuration

The shoulder configuration of a cylindrical roller bearing shall provide proper roller guidance, minimize roller end wear, and prevent stress concentrations.

The corner configuration employing grind relief (fig. 22(a)) is preferred because this design eliminates possible interference of the radiused corner with roller corner radius (r, fig. 21).

The rake angle α should be 0° 15′ to 0° 45′.

The shoulder height h should be 14 to 20 percent of the roller diameter to provide proper guidance.

Maximum axial grind relief A (fig. 22(a)) should be less than roller end radius breakout a in figure 21.

The maximum radial grind relief B (fig. 22(a)) should be less than the corner radius breakout height b in figure 21.

3.2.3 Cage Design

3.2.3.1 Cage Strength

The cage design shall possess sufficient strength and rigidity to maintain rolling element spacing and to withstand hoop stress from centrifugal force.

A one-piece cage design should be specified whenever possible. A supporting structure should be provided for all cages made of nonmetallic materials except those having fabric reinforcement (i.e., glass-fabric-supported PTFE (Armalon) and linen-supported phenolic). The supporting structure should be a full metal shroud or reinforced band design (fig. 26). The shroud material must be undercut to prevent contact with rolling elements or race land. A supporting structure for two-piece cages made of Armalon should be provided. The supporting structure can be full metal shrouds for inner-land-riding cages or riveted side bands for outer-land-riding cages.

Minimum allowable radial thickness must be determined individually for each application and cage material. A suggested minimum radial thickness dimension for cages made of unsupported nonmetallic materials is 0.090 in. A suggested minimum ratio of thickness to mean diameter is 0.015 in. Thinner sections should be tested prior to any use in production turbomachinery.

3.2.3.2 Coolant Flow Area

The cage shall allow free flow of coolant through the bearings.

Free through-flow of coolant can be assured by providing one of the following design features:

(1) Outer-land-riding cage with slim cross section (fig. 27(a)).
(2) Inner-land-riding cage with coolant flow cutouts (ref. 9) (fig. 27(b)).
(3) Outer-land-riding cage with flow cutouts (fig. 27(c) and (d)).

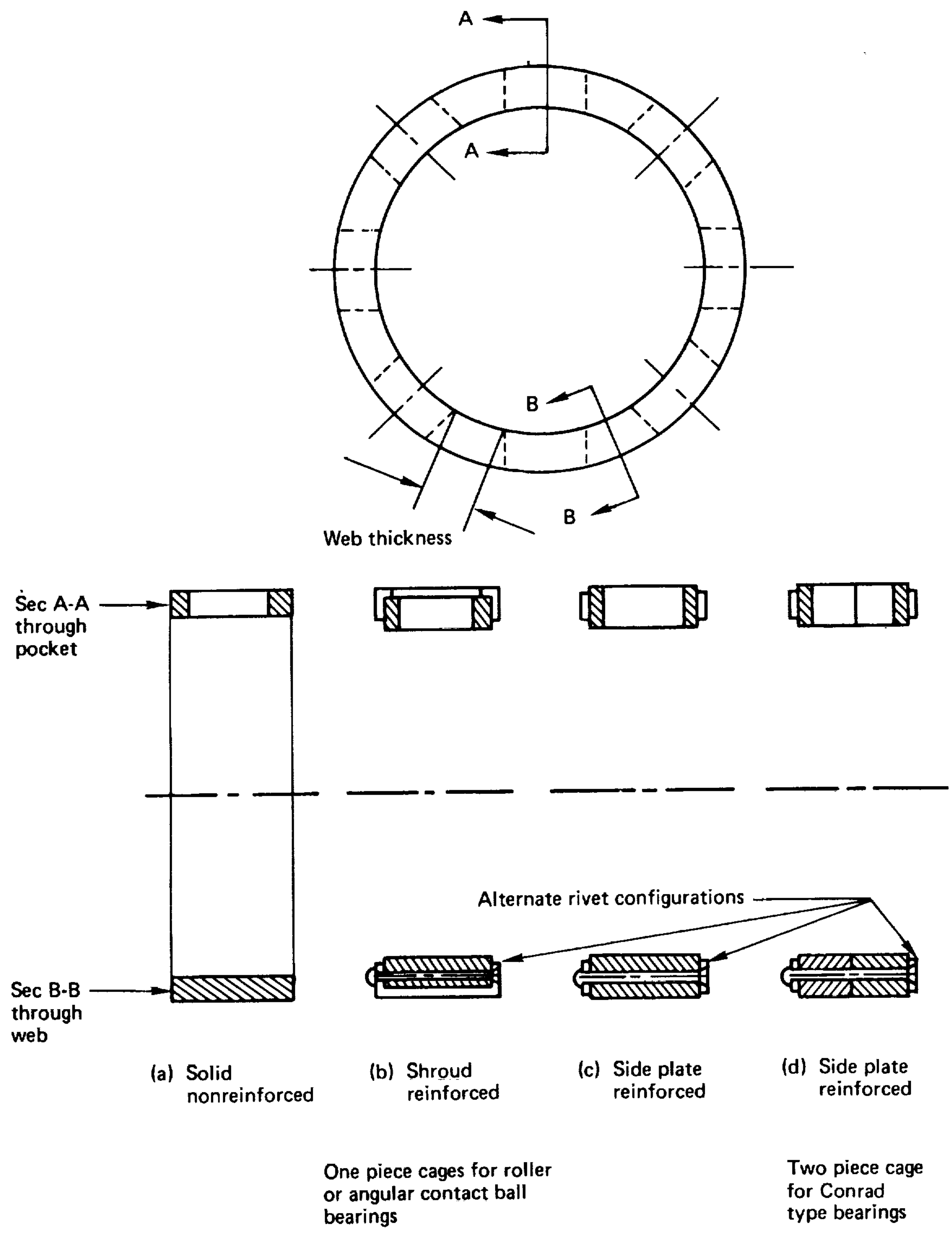
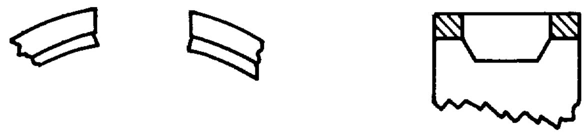
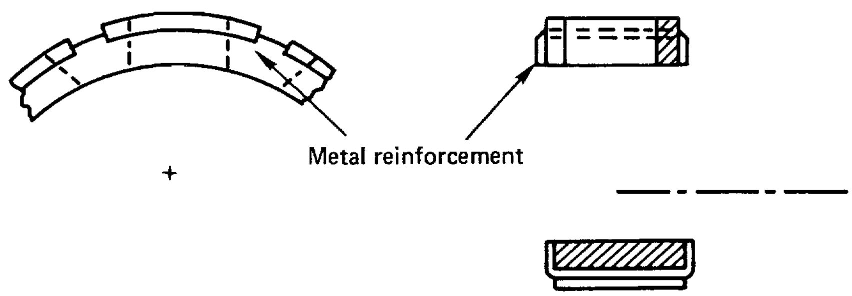


Figure 26.—Cage construction.



(e) Ball retaining cage

(f) Roller retaining cage



(g) Outer land riding shrouded cage (ref. 1)

Figure 26.—Concluded. Cage construction.

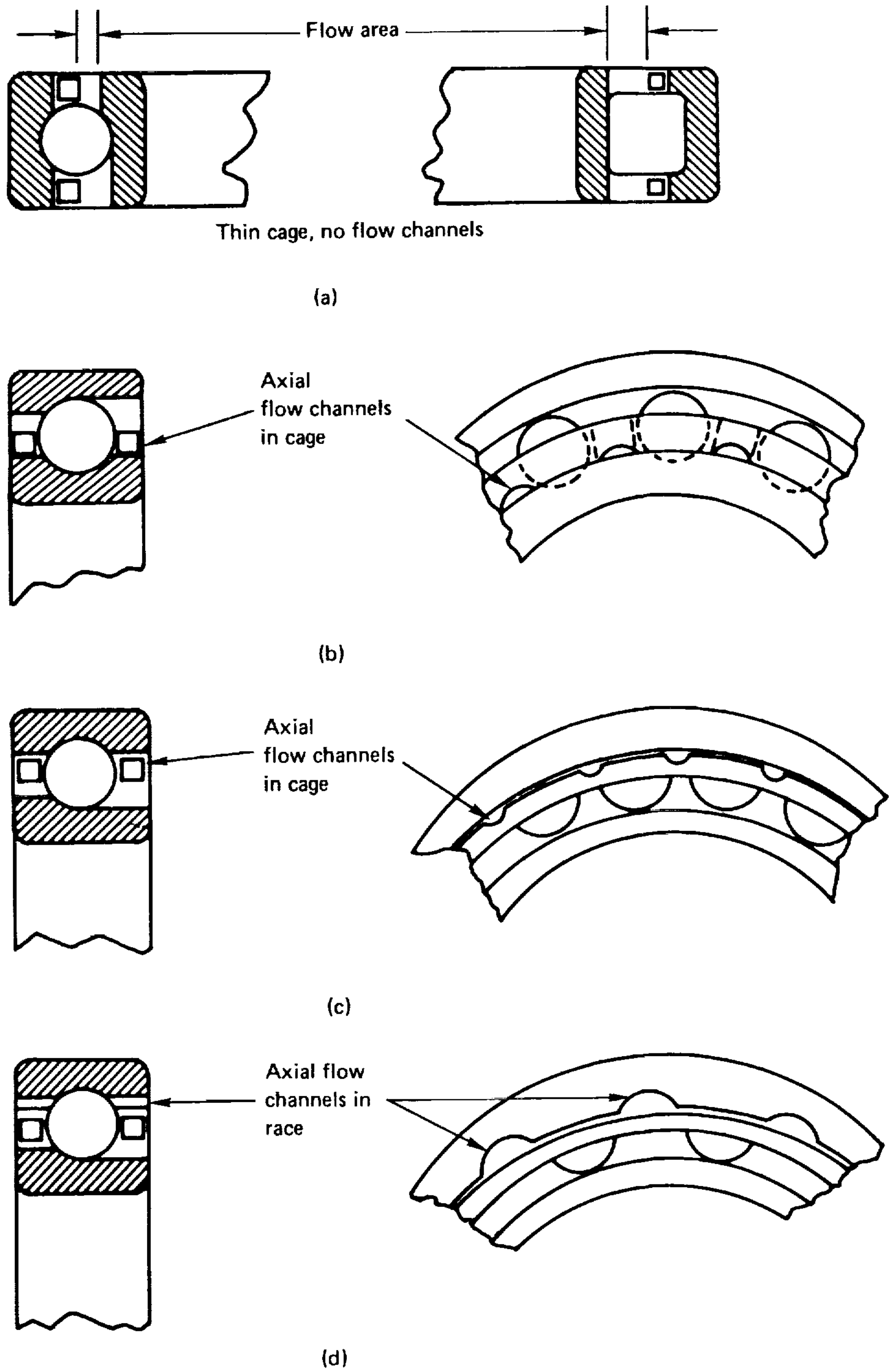


Figure 27.—Coolant flow channels.

3.2.3.3 Cage-Rolling Element Contact

The cage pocket surface shall meet the rolling element so that the rolling elements do not tend to "climb over" or deform the cage web.

The cage pocket surface should meet the rolling element at its pitch diameter to avoid a wedging action of the rolling element that could deform the cage.

3.2.3.4 Web Thickness

The cage web shall be thick enough to provide adequate strength with allowances for wear.

The following guides (see fig. 26) should be used where test results are not available:

(1) Minimum cage web thickness for metallic cages should be 0.070 in.
(2) Minimum cage web thickness for nonmetallic cages should be 0.150 in. (ref. 9).

3.2.3.5 Guiding Land Clearance

Cage guiding land clearance shall be adequate at all operating temperatures and rotating speeds.

A minimum cage guiding land clearance of 0.003 in./in. of guiding land diameter under all operating conditions should be provided; allowances for differential thermal growth of cage and race material and for centrifugal growth of race and cage material should be included.

3.2.3.6 Pocket Clearance

Cage pocket clearance in ball bearings shall be sufficient to allow ball advance and retard due to combined load or misalignment. Roller bearing cage pocket clearances shall not become negative.

Testing may be required to select the optimum clearance. The following can be used as guidelines:

(1) Cage pocket clearance for ball bearings under combined axial and radial load or misalignment should be 0.035 in./in. of ball diameter.
(2) Cage pocket clearance for ball bearings under normal alignment and thrust only or radial only should be 0.025 in./in. of ball diameter.

(3) Cage pocket clearance for roller bearings: 0.025 in./in. roller diameter (circumferential); 0.005 in./in. of roller length (axial).
(4) Cage half register in a Conrad-type bearing should be sufficiently accurate that cage pocket clearance is maintained.
(5) Elliptical cage pockets should be used to reduce axial cage clearance and still obtain adequate circumferential clearance.

3.2.3.7 Cage Sides Protrusion

The cage side shall not make contact with components adjacent to the bearing.

A tolerance stackup of cage position should be performed allowing for extremes of ball dynamic position, cage clearances, and cage dimensions. If cage protrusion occurs, either ensure that no components can come in contact with the cage by specifying their position or minimize protrusion by using countersunk rivet heads or reducing cage width. If cage width is reduced, ensure that the cage structure is not unduly weakened.

3.2.3.8 Cage Rivet

A cage rivet shall provide positive tension throughout the design temperature range without overstressing the cage material.

To size the rivet diameter and length, take into account the compressive strength of cage material, the tensile strength of the rivet material, and the elastic moduli, Poisson's ratio, and thermal expansion properties for both materials. Appropriate analyses should demonstrate that under all operating conditions the rivet will provide a positive tension without yielding the cage material.

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GLOSSARY

MATERIAL DESIGNATIONS

Common designation as used in Monograph	Specification or other identification
Armalon	Glass-fabric-supported PTFE (Teflon). Normal form for bearing cages in mandrel wrapped tubing. Trade mark of E. I. duPont. To be fully defined, the density, size of weave, and mechanical properties must be specified.
AMS 4616	Silicon-bronze alloy.
Bower 315	Alloy steel used for high-temperature roller bearings. Composition is specified by Bower Roller Bearing Division of Federal-Mogul-Bower, Inc., Detroit, Michigan.
BN 440	Commercial beryllium-nickel alloy made by Kawecki Berylco Industries, Inc.
CRES	Corrosion resistant steel.
EDA	Ethylene diamine.
GFT	Glass-filled Teflon: PTFE with random glass fibre content, specified in percent by volume.
Halmo	Alloy steel used for high-temperature bearings by New Departure-Hyatt Division of General Motors Corporation.
Haynes 25	Cobalt-chromium-nickel alloy covered under AMS 5796.
IRFNA	Inhibited red fuming nitric acid.
K-Monel	Nickel-copper alloy conforming to AMS 4676 and Federal Specification QQ-N-286, Class A annealed.
K-5H	Sintered tungsten—titanium carbide material (Kennametal, Inc.).
K-96	Sintered tungsten-carbide material with cobalt binder manufactured by Kennametal, Inc., Latrobe, Pennsylvania.
K-162B	Sintered titanium-carbide material with nickel-molybdenum binder (Kennametal, Inc.).

M-10	AISI designation for molybdenum-steel alloys.
M-50	AISI designation for molybdenum-steel alloys.
PTFE	Chemical composition: Polytetrafluoroethylene. Commercial name: Teflon.
Rulon A	PTFE (Teflon) with random glass fibre content 15 percent by volume. T. F. Dixon Company, Bristol, Rhode Island.
Salox M	Commercial designation for PTFE filled with 40 percent bronze powder.
S-Monel	Nickel-copper alloy conforming to Navy Specification 46N7, Class B.
Stellite	Commercial cobalt-chromium-nickel alloys manufactured by Haynes Stellite Corporation.
UDMH	Unsymmetrical dimethyl hydrazine.
50-50	Mixture: 50 percent N_2H_4-50 percent UDMH.
4130	AISI designation for low alloy carbon steel.
6414	Premium quality AISI 4340 steel.

ORGANIZATIONS

Abbreviation	Identification
ABEC	Annular Bearing Engineering Committee of the AFBMA. ABEC-1, -3, -5, -7, -9 are progressively more precise tolerance specifications for ball bearings.
AFBMA	Anti-friction Bearing Manufacturers Association.
AISI	American Iron and Steel Institute.
AMS	Aerospace Material Specifications published by SAE.
ASA	American Standards Association.
RBEC	Roller Bearing Engineering Committee of the AFBMA. RBEC-1, -5, etc. are progressively more precise tolerance specifications for roller bearings.
SAE	Society of Automotive Engineers.

MISCELLANEOUS TERMINOLOGY

Term or symbol	Definition
AA	Arithmetic average.
$\beta_i - \beta_o$	Difference between inner race contact angle β_i and outer race contact angle β_o that occurs under dynamic conditions.
B_{10}	Expression for bearing fatigue life defined as the operating time that 90 percent of identical bearings will exceed without fatigue failure.
BSV	Ball speed variation.
C_c	Clearance reduction due to centrifugal growth of inner race.
C_d	Unfitted diametral clearance.
C_f	Clearance change due to press fitting.
C_o	Operating diametral clearance that should be selected in the original design.
C_t	Clearance change caused by temperature difference between races.
circumferential	Lying in a plane perpendicular to the shaft centerline.
D_{avg}	Average raceway diameter.
DN	Product of bearing bore in millimeters and shaft speed in revolutions per minute.
d	Diameter of ball or roller.
E	Pitch diameter of ball complement.
EHD	Elastohydrodynamic.
F_i	Inner transverse race curvature: inner race radius divided by ball diameter.

F_o	Outer transverse race curvature: outer race radius divided by ball diameter.
$F_o + F_i - 1$	Expression for total bearing curvature.
FLOX	Fluorine-oxygen mixture.
IRC	Inner race control.
NVR	Nonvolatile residue.
ORC	Outer race control.
R_c	Hardness on the Rockwell C scale.
TIR	Total indicated runout.
t_{avg}	Average cross-section radial thickness of race.
transverse	Lying in a plane containing the shaft centerline.
windage	Circulation or pumping of fluids caused by impeller action of rotating components. That is, a gas contained in a cavity with shafts, gears, or bearings will circulate within the cavity by windage.

NASA SPACE VEHICLE DESIGN CRITERIA MONOGRAPHS ISSUED TO DATE

ENVIRONMENT

SP-8005	Solar Electromagnetic Radiation, June 1965
SP-8010	Models of Mars Atmosphere (1967), May 1968
SP-8011	Models of Venus Atmosphere (1968), December 1968
SP-8013	Meteoroid Environment Model—1969 (Near Earth to Lunar Surface), March 1969
SP-8017	Magnetic Fields—Earth and Extraterrestrial, March 1969
SP-8020	Mars Surface Models (1968), May 1969
SP-8021	Models of Earth's Atmosphere (120 to 1000 km), May 1969
SP-8023	Lunar Surface Models, May 1969
SP-8037	Assessment and Control of Spacecraft Magnetic Fields, September 1970

STRUCTURES

SP-8001	Buffeting During Atmospheric Ascent, revised November 1970
SP-8002	Flight-Loads Measurements During Launch and Exit, December 1964
SP-8003	Flutter, Buzz, and Divergence, July 1964
SP-8004	Panel Flutter, May 1965
SP-8006	Local Steady Aerodynamic Loads During Launch and Exit, May 1965
SP-8007	Buckling of Thin-Walled Circular Cylinders, revised August 1968
SP-8008	Prelaunch Ground Wind Loads, November 1965

SP-8009	Propellant Slosh Loads, August 1968
SP-8012	Natural Vibration Modal Analysis, September 1968
SP-8014	Entry Thermal Protection, August 1968
SP-8019	Buckling of Thin-Walled Truncated Cones, September 1968
SP-8029	Aerodynamic and Rocket-Exhaust Heating During Launch and Ascent, May 1969
SP-8031	Slosh Suppression, May 1969
SP-8032	Buckling of Thin-Walled Doubly Curved Shells, August 1969
SP-8035	Wind Loads During Ascent, June 1970
SP-8040	Fracture Control of Metallic Pressure Vessels, May 1970
SP-8046	Landing Impact Attenuation For Non-Surface-Planing Landers, April 1970

GUIDANCE AND CONTROL

SP-8015	Guidance and Navigation for Entry Vehicles, November 1968
SP-8016	Effects of Structural Flexibility on Spacecraft Control Systems, April 1969
SP-8018	Spacecraft Magnetic Torques, March 1969
SP-8024	Spacecraft Gravitational Torques, May 1969
SP-8026	Spacecraft Star Trackers, July 1970
SP-8027	Spacecraft Radiation Torques, October 1969
SP-8028	Entry Vehicle Control, November 1969
SP-8033	Spacecraft Earth Horizon Sensors, December 1969
SP-8034	Spacecraft Mass Expulsion Torques, December 1969

SP-8036	Effects of Structural Flexibility on Launch Vehicle Control Systems, February 1970
SP-8047	Spacecraft Sun Sensors, June 1970

CHEMICAL PROPULSION

SP-8025	Solid Rocket Motor Metal Cases, April 1970
SP-8041	Captive-Fired Testing of Solid Rocket Motors, March 1971
SP-8051	Solid Rocket Motor Igniters, March 1971

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